NUMERICAL INVESTIGATION ON THE PERFORMANCE OF INERTANCE TUBE PULSE TUBE REFRIGERATOR VARYING COMPRESSOR FREQUENCY

National Institute of Technology, Rourkela

In partial fulfilment of the requirement for the degree of

Master of Technology

In

Mechanical Engineering

(Cryogenic & Vacuum Technology)

By

B. Mohan Kumar

(Roll No. - 212me5446)

Under the guidance of

Dr. Suman Ghosh

Department of Mechanical Engineering
National Institute of Technology Rourkela
Rourkela-769008 (India)

Dedicated to

My Beloved Parents,

Dushmanta Behera

&

Bharati Behera

National Institute of Technology
Rourkela

CERTIFICATE

This is to certify that the research work that has been presented in this thesis entitled "**NUMERICAL INVESTIGATION ON THE PERFORMANCE OF INERTANCE TUBE PULSE TUBE REFRIGERATOR VARYING COMPRESSOR FREQUENCY**" by **B. Mohan Kumar (Roll No. 212ME5446)** has been carried out under my supervision in partial fulfilment of the requirements for the degree of Master of Technology in Mechanical Engineering (Cryogenics and Vacuum Technology Specialization) during session 2012-2014 in the Department of Mechanical Engineering, National Institute of Technology, Rourkela.

To the best of my knowledge, this dissertation work has not been submitted in any other college or university at any time prior to this, for the award of any degree or diploma.

Place: NIT Rourkela, Rourkela-769008, Orissa. Date: 2nd June, 2014

Dr. Suman Ghosh
Assistant Professor
Department of Mechanical Engineering
NIT Rourkela, Rourkela-769008, India

DECLARATION

I certify that

a. The work contained in the thesis is original and has been done by myself under the general supervision of my supervisor.
b. The work has not been submitted to any other Institute for any degree or diploma.
c. I have followed the guidelines provided by the Institute in writing the thesis.
d. I have conformed to the norms and guidelines given in the Ethical Code of Conduct of the Institute.
e. Whenever I have used materials (data, theoretical analysis, and text) from other sources, I have given due credit to them by citing them in the text of the thesis and giving their details in the references.
f. Whenever I have quoted written materials from other sources, I have put them under quotation marks and given due credit to the sources by citing them and giving required details in the references.

Signature of the Student

ACKNOWLEDGEMENT

I express my deep sense of gratitude towards my thesis supervisor Dr. Suman Ghosh for providing their unwavering support, encouragement and consummate feedback throughout the course of this study and in the participation of this thesis. Without his illustrious guidance this thesis would not have seen the light of the day.

I express my heartiest thanks to Professor **K.P. MAITY, HOD, Department of Mechanical Engineering**, and also the other staff members of NIT Rourkela for providing me essential facilities that is required to conduct the research and complete my thesis.

I need to further thank all my friends and special thanks to Tekendra Bhatt, Kumar Samal & Deepak Bhunya who always took the time to listen, even when I was just complaining.

Date: 2nd June, 2014 B. MOHAN KUMAR

CONTENTS

LIST OF SYMBOLS AND ABBREVIATIONS

Symbols	Description
γ	Porosity
∇	Gradient operator
ρ	Density of the gas
$\vec{v}$	Velocity in vector form
r	Radial coordinate
x	Axial coordinate
v_r	Velocity in radial coordinate
v_x	Velocity in axial coordinate
p	Static pressure
τ	Stress Tensors
$\vec{g}$	Acceleration due to gravity
τ	Fluid molecular viscosity
I	Unit (Identity) tensor
K	Gas thermal conductivity
k_t	Turbulence thermal conductivity
C_P	Specific heat of gas
T	Temperature of the gas
V	Local velocity
$\overrightarrow{\Psi_J}$	Diffusion flux of species
R	Gas constant
μ	Dynamic viscosity of fluid
α	Permeability
C	Inertial resistance factor
$\vec{v}$	Velocity
K_S	Thermal conductivity in solid phase
K_f	Thermal conductivity in solid phase
k_{eff}	Effective thermal conductivity
H	Enthalpy flow
P_d	Pressure in dynamic condition

V	Volume flow-rate
X	Piston displacement
X_a	Piston displacement amplitude
ω	Angular Frequency
F	Frequency in Hertz
t	Time period in second

ABBREVIATION

PTR	Pulse tube refrigerator
BPTR	Basic pulse tube refrigerator
OPTR	Orifice type pulse tube refrigerator
DIPTR	Double inlet pulse tube refrigerator
ITPTR	Inertance tube pulse tube refrigerator
CHX	Cold heat exchanger
HHX	Hot heat exchanger
CFD	Computational fluid dynamics
FOM	Figure of Merit
UDF	User defined function
FVM	Finite Volume Method

LIST OF TABLES

LIST OF FIGURES

Abstract

Numerical studies of Pulse Tube Refrigeration are performed in the present dissertation to investigate the influence compressor frequency on the **performance of Stirling type single stage Inertance Tube Pulse Tube Refrigerator (ITPTR)**. The entire inertance tube pulse tube refrigerator (ITPTR) operating under a specified boundary conditions are modelled using Finite Volume Method (FVM) with commercial software package ANSYS with Dynamic Meshing and User Defined Function. It is found that the performance of the ITPTR and the cooling rate is strongly dependent on the frequency of the compressor. The frequency study have been performed for two different dimensions of the ITPTR separately.

The frequency of the compressor has been varied from 10 Hz to 100 Hz for ITPTR with dimension-1. The rate of cooling is found maximum when the frequency of the compressor is used as 20 Hz for ITPTR with dimension-1. For the ITPTR with dimension-2, the frequencies have been varied from 20 Hz to 34 Hz. The rate of cooling is found maximum when the frequency of the compressor is used as 34 Hz for ITPTR with dimension-2.

Keyword: Inertance Tube Pulse Tube Refrigerator, Compressor, Cold Heat Exchanger, Regenerator, Hot heat exchanger, Pulse Tube,Frequency, Finite Volume Method (FVM)

Chapter - 1

Introduction
&
Literature Review

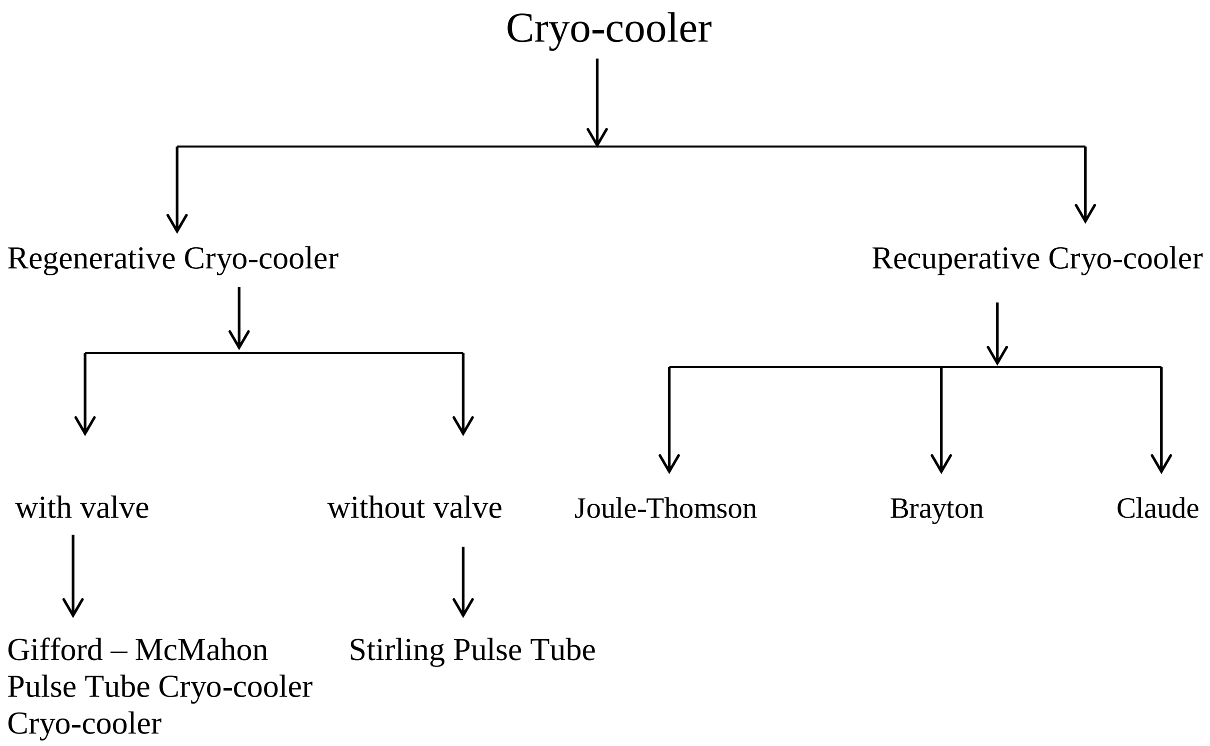
This chapter contains basic introduction of cryogenics, Organisation of Cryo-coolers, Uses of Cryo-coolers and Introduction to Pulse Tube Refrigerator (PTR), working of PTR as well as its advantages over Cryo-cooler, Application of PTR andclassification of PTR in different ways, Literature survey for this current study, gaps in literatures and the objective for current research.

1.1 INTRODUCTION:

Cryogenic comes from Greek words “Kruoz” and “genes” which means “freezing” and “to produce” respectively & is introduced by Kammerlingh Onnes in 1908. Cryogenics is the technology that associated with very low temperature. In engineering, the temperature range for cryogenics can be described in between absolute zero to 123K.

Cryo-coolers are the refrigeration machines which can operate a temperature of less than 123K and with a small refrigeration capacity. Basically two types of cryo-coolers are uses in the field of cryogenics such as recuperative cryo-coolers and regenerative cryo-coolers.

1.1.1 Organisation of Cryo-coolers:



1.1.2 Uses of Cryo-coolers:

Cryo-coolers has a wide range of application as summarized below,

a) It has a wide range of application in military appliances.
b) It has a wide range of application in environmental pollution.
c) It has a wide range of application in medical appliances.
d) It has a wide range of application in transportation of vehicles.
e) It has a wide range of application in police & security appliances.
f) It has a wide range of application in agriculture & Biology.
g) It has a wide range of application in commercial uses.

Due to the above applications of cryo-coolers, high concert efficiency, long life time, low vibration, reliability, lesser size besides weight become essential aspect for development of cryo-coolers. Stirling coolers & G-M type coolers have been widely used in different fields with different application because of its low heat transfer losses. In both the cryo-coolers (Stirling type and G-M type) there are moving parts in the cold ends. The PTR is efficient in comparison to other cryo-coolers due to its easier construction and no moving parts at the cold end. Now days for these various reasons more consideration has been focused on research on PTR in recent years.

1.1.3 Pulse Tube Refrigerators:

Pulse tube refrigerators (PTR) are a hollow tube in which cooling effect at one end & pulsating pressure at other end. It is a class of cryo-coolers which have capable of reaching a cryogenic temperature operated without a moving part at their low temperature end. PTR plays an important role in the ground of cryogenic refrigeration because of its various advantages. The main advantages of this device, as compared to Stirling & Gifford-McMahon system, are its trustworthiness and long life due to absence of moving parts at low temperature region.

1.1.4 Working Principle of the Pulse Tube Refrigerators:

PTR is a device having capable of cooling to temperature lower than 123K. PTR outfits the concept of oscillatory expansion and compression of the gas inside a closed volume to accomplish desire cooling. PTR is an unsteady system that entails time dependent solution. A PTR is a closed system that uses a fluctuating pressure at one side to produce a fluctuating gas flow in the remaining part of the system.

1.1.5 Application of PTR:

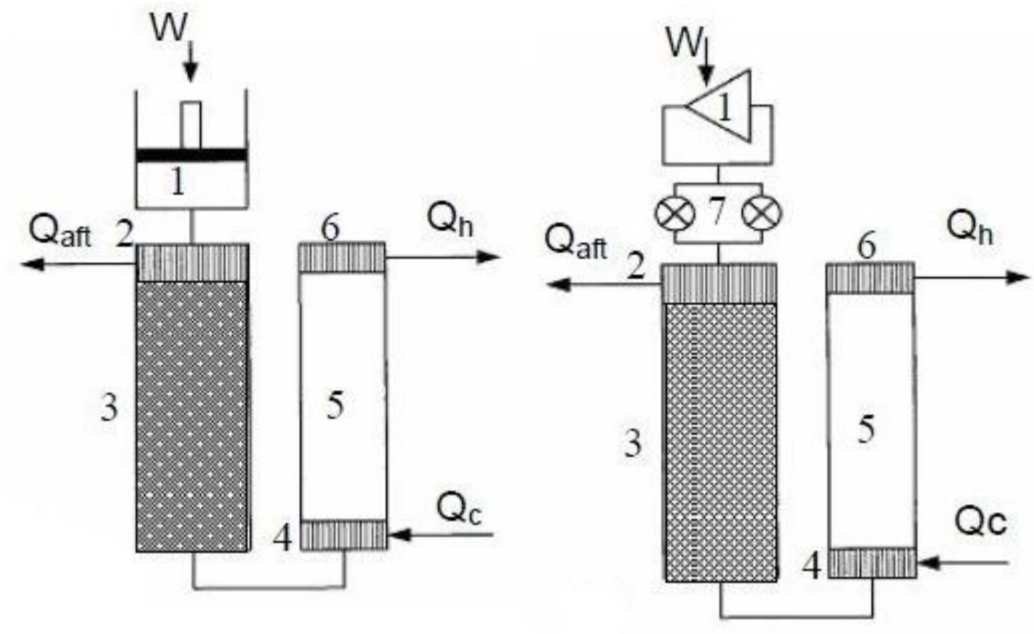
- It can be used for liquefying oxygen on MARS.
- These are uses for freezing the astronomical detectors which uses liquid cryogens.
- The PTRs are used for freezing the infrared sensors in the missile.
- The PTRs are used in satellite.
- These are uses in the cooling of superconductors and semiconductors.
- These are uses in other applications such as in cryo-pumps, cooling of radioactivity guards, liquefying natural gases, SQUID, Magnetometers and Superconducting Magnets, small electromagnetic interference etc.

1.1.6 Organization of Pulse Tube Refrigerators:

We can classify PTR in different ways as below,

- On the basis ofnature,
  - Stirling type PTR
  - Gifford-McMahon (G-M) type PTR.
- On the basis of development,
  - Basic PTR
  - Orifice PTR
  - Multiple Inlet PTR
  - Double Inlet PTR
  - Inertance Tube PTR
  - Single Stage PTR
  - Thermo-acoustic PTR
  - Multi Stage PTR
- On the basis of geometry or Shape,
  - In-line type PTR
  - Coaxial type PTR
  - U type PTR

The overview of PTR is shown below where as 1.1(a) shows the overview of Stirling type PTR & 1.1(b) shows the overview of G-M type PTR.



1.1(a) 1.1(b)

Figure 1.1: Structural representation of Stirling type PTR & a G-M type PTR (Banjare et al., 2007)

Different components of the above figure as follows:

1 - Compressor,

2 - After-cooler,

3 - Regenerator,

4 - Cold Heat-exchanger,

5 – Pulse-tube,

6 - Warm Heat-exchanger,

7 - Valve

Fig.1.2 demonstrates schematic representation of Stirling type BPTR. It contains the following components such as compressor, after cooler, regenerator, cold heat-exchanger, pulse tube and hot heat-exchanger. It was developed in 1964 by Gifford & Longsworth. BPTR has reached the temperature at the cold end about 124K.

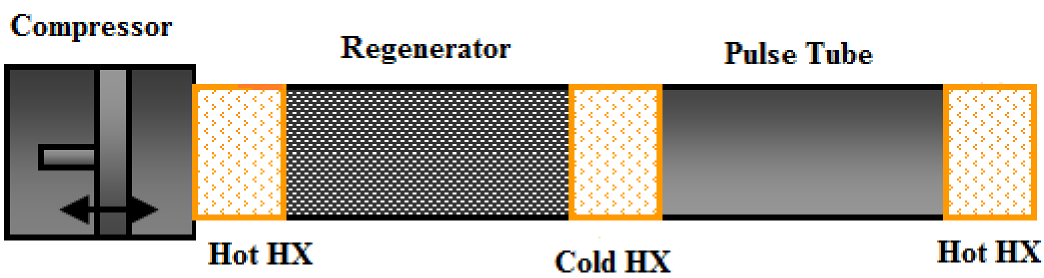


Figure 1.2: Stirling type Basic Pulse Tube Refrigerator (Etaati, 2007)

Fig.1.3 demonstrates the schematic representation of Stirling type OPTR. It contains the following components such as compressor, after cooler, cold heat-exchanger, regenerator, pulse tube and warm heat-exchanger, orifice, reservoir. It was developed in 1964 by Mikulin. It is much efficient than BPTR.

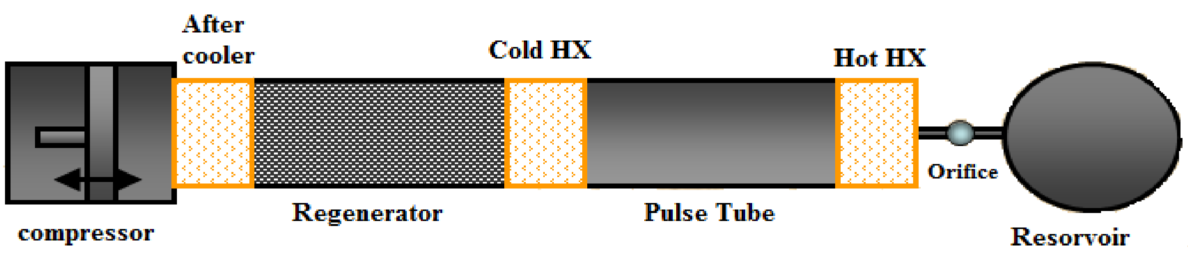


Figure1.3: Stirling type Orifice Pulse Tube Refrigerator (Etaati, 2007)

Fig.1.4(a) & Fig.1.4 (b) demonstrates the schematic representation of Stirling type double-inlet pulse tube refrigerator (DIPTR) G-M type DIPTR respectively. It contains the following components such as compressor, after cooler, cold heat-exchanger, pulse tube, regenerator and hot heat-exchanger, orifice, double inlet valve, reservoir. It is more efficient than OPTR.

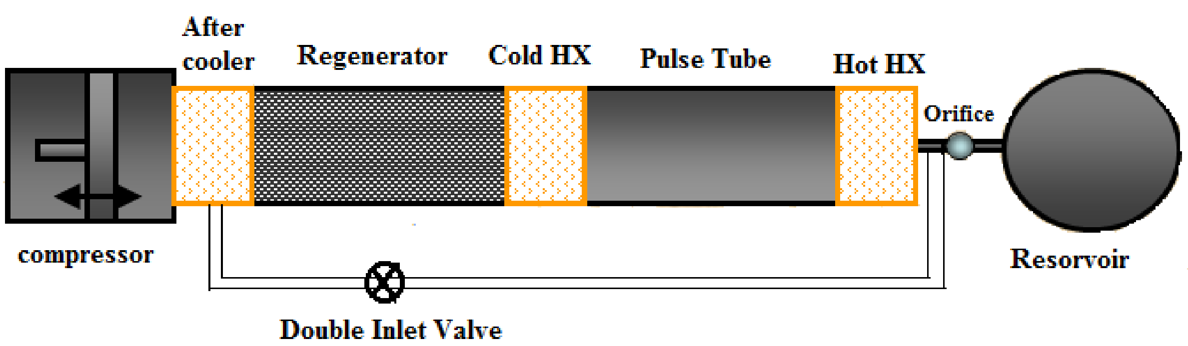


Figure1.4(a): Stirling type DIPTR(Etaati, 2007)

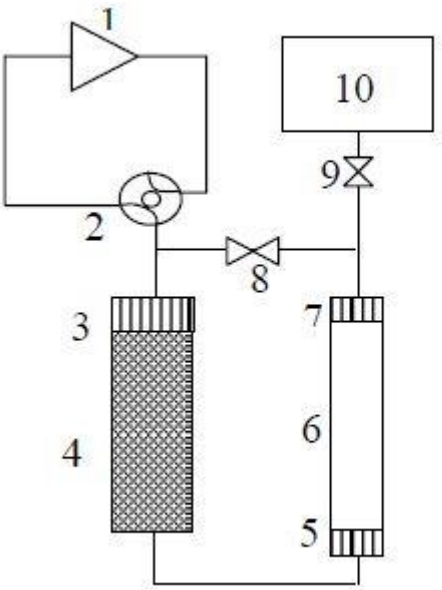


Figure1.4(b): G-M type Double Inlet Pulse Tube Refrigerator
(Banjare et al., 2007)

Where,

1 - Compressor,

2 - Rotary valve,

3 - Regenerator hot end,

4 - Regenerator,

5 - Cold heat exchanger,

6 - Pulse tube,

7 - Hot heat exchanger,

8 - Double Inlet valve,

9 - Orifice valve,

10 - Reservoir

Fig.1.5 demonstrates the graphic representation of Stirling type Inertance Tube PTR which is the recent PTR & invented by Radebaugh in 1994. Here the orifice valve is exchanged by a much extended inertance tube consuming very small inside diameter. It contains the following components such as compressor, after cooler, cold heat-exchanger, regenerator, pulse tube and hot heat-exchanger, orifice, reservoir. It is more efficient than all other PTR.

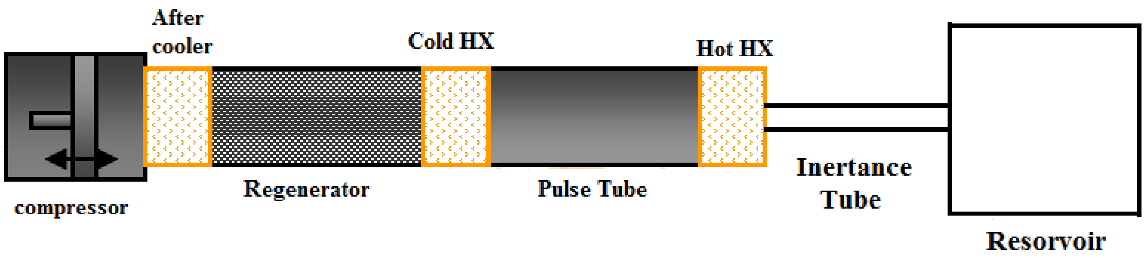


Figure1.5: Stirling type ITPTR (Etaati, 2007)

1.2 Literature Survey:

One of the most main components of most cryogenic plants is the pulse tube refrigerator. Due to Its extensive practical application, the pulse tube refrigerator has attracted the attention of a large number of researchers over the years. An investigation involving experimental as well as theoretical studies has been reported in literature.

Journals such as cryogenics and major conference proceeding of the International Cryogenic Engineering conference devote a sizable portion of their contents to research findings on PTR.Several studies have been undertaken in last few decades on the PTR. A few of the studies regarding PTR described below,

In 1964 Prof. W. Gifford & R. Longsworth replace the displacer establish in Stirling & Gifford McMahon refrigerators by a hollow tube which is called as pulse tube & the system is called as Basic PTR. Previously, 150 K was achieved in single stage PTR & 120 K in a two stage PTR. Mikulin et al. (1984) disclosed that Pulse tube efficiency could be improved by clasp a surge tank at the right side, through an orifice. Zhou et al. (1988) exposed to grow the specific refrigeration power over the regenerator. Radebaugh et al. (1986) have done work with three kinds of PTRs. In that work, they got a recordable lower temperature of 60 K by means of a one stage pulse tube. Huang et al. (1992) investigate the performance characteristics of PTRs by experimental study. Wu et al. (1994) performed a mathematical study on OPTR by means of the technique of features and made an introductory judgement with experimentation. Xu et al. (1999) performed a theoretical investigation and got a temperature of almost 2K. Based on the previous study Liang et al. (1996) established the compound-pulse-tube model and combined the thermal and viscous effect of the wall and anticipated a thermal layer in the pulse tube. Charles et al. (1999) validate the properties of the stable flow in the double inlet configuration by performing an experiment at CEA/SBT on PTR. De Boer et al. (2002) developed a model of BPTR by improving various factors like gas motion with cooling and heating effect which results more accurate temperature profile and they analyse the BPTR having regenerator and heat exchangers at different ends. Cha et al. (2006) demonstrated Inertance tube PTR working under various thermal boundary conditions & different dimensions. Antao et al. (2011) reported the important flow and heat transfer process found in an orifice type PTR. Zhang et al. (2007) have analysed the properties of the reservoir on the basis of thermodynamic performance of different constituents of anOPTR and a double-inlet PTR by uniting a linearized model. Conrad et al. (2008) studied the effect of pulse tube diameter scaled to a non-dimensional value by relevant boundary layer thickness; on auditory streaming in the pulse tube was performed using CFD modelling. Farouk et al. (2012) investigate the flow and the methods in a co-axial type single stage OPTR. Wang et al. (1992) improved the performance of OPTR by friction of gas flow, heat transfer in the heat exchanger, regenerator & different properties of material. Gan et al. (2000) investigates on multi-phase helium and nitrogen mixtures in a single-stage PTR. Here they got same temperature in compare with the model having working fluid only helium, so it is more efficient. Yang et al. (2005) designed a two-stage Stirling-type U-shape pulse tube cryo-cooler operated by a 10 kW linear compressor. Gao et al. (2000) improve the performance of a single-stage PTR, gas mixtures are used for comparison, which have been used in other PTR. Zhang et al. (2007) implements a 2D computational fluid dynamic (CFD) simulation of a GM-type OPTR. Moldenhauer et al. (2013) studied the pressurized laboratory scale PTR by means of designing experimental tools. Antao et al. (2013) studied the characteristics concert of the OPTR for various values of the mean pressure of helium (3.5 Bar to 22 Bar), different values of operating frequencies and operation sizes of orifice introductory.

1.3 Gaps in Literature:

From the extensive literature survey discussion it is found that though several attempt on PTR have so far been but still there are some parts are missing so further research is required.

- Some of the gaps in literature are as follows:
  - Maximum work carried out is based on experimental basis.
  - Experimental study of PTR is very costly and time taking.
  - Change of materials or dimension in experimental analysis are also not so easy.

1.4 Aims and Objective:

From the Literature survey conclusion can be drawn that a few efforts have been done to investigate the suitability for simulation of Stirling type single stage ITPTR by using CFD.

Numerical investigations on performance of ITPTR with varying the compressor frequency by using helium as working fluid.

1.5 Organization of the Thesis:

The thesis is organised into six different chapters. First chapter contains introduction of PTR and various types of PTR. Brief history of PTR described on the review of earlier works gaps in literatures & scope of present work has been described. Second chapter describes the problem description in details. Third chapter contains the basic methodology used for the modelling by using finite volume method (FVM) and the dynamic meshing of the structure by using my own user define function (UDF). Working of PTR, governing equations associated with PTR and Porous zone defined for few parts of the PTR are also described here. Fourth chapter contains the result & discussion of the problem described in second chapter. Here two models of PTR are described with two different shape of pulse tube having same boundary condition with varying the compressor frequency and it also shows the comparison among different temperature we got from different frequency. Fifth chapter contains the conclusion on the basis of result & discussion described in fourth chapter. Sixth chapter contains the future areas of research on PTR.

Chapter-2

PROBLEM DESCRIPTION

Present chapter describe the details formulation the problem to meets the aims and objectives defined in chapter 1. The shape of ITPTR on which simulation is carried out is shown in Figure 2.1 Compressor uses to creating a harmonic oscillation for the gas inside the system. **After cooler** uses to remove the heat from the compressor during the compression process. **Regenerator** is a matrix as a porous media having high heat capacity and low conductivity to exchange the heat with the gas. It absorbs heat from the entering gas through the forward stroke of the compressor and delivers that heat back to the gas through the back stroke. **Cold heat exchanger** absorbs the heat of the environment because of cooling down in the expansion cycle. **Pulse Tube** is a hollow tube in which cooling effect at one end & pulsating pressure at other end use to pumping the heat from cold heat-exchanger. **Warm heat-exchanger** releases the heat created in the compression cycle to the environment. **Inertance tube** is a connector between hot heat exchanger & reservoir. It is very long in size & having very small diameter & it is an inlet for the flow resistance. **Reservoir** is having much more volume in compare with rest of the system& from the wall of reservoir gas again starts to move back in order to produce cooling effect. Different cases for which the numerical study has been made are shown below point wise.

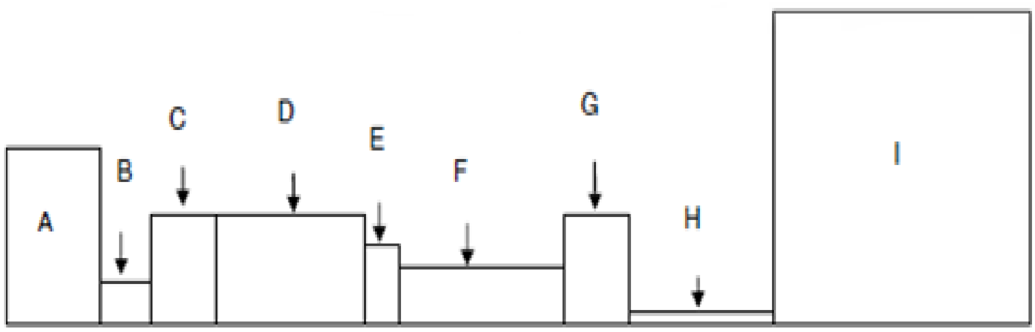


Figure 2.1: Simulated ITPTR system (Cha et al. 2006)

2.1 FREQUENCY STUDY WITH DIMENSION – 1:

- Different dimensions taken for the simulation are as follows,

Components	Radius (m)	Length (m)
Compressor(A)	0.00954	0.0075
Transfer line (B)	0.00155	0.101
After Cooler (C)	0.004	0.02
Regenerator (D)	0.01	0.025
Cold-heat exchanger (E)	0.003	0.0057
Pulse-tube (F)	0.0075	0.03
Warm-heat-exchanger (G)	0.004	0.01
Inertance-tube (H)	0.000596	1.07
Surge volume (I)	0.013	0.13

(Table 1.1)

2.1.1 Dimension – 1 with frequency 10 Hz:

Here the piston of the compressor engaged for the simulation moving with frequency 10 Hz.

2.1.2 Dimension – 1 with frequency 20 Hz:

Here the piston of the compressor engaged for the simulation moving with frequency 20 Hz.

2.1.3 Dimension – 1 with frequency 60 Hz:

Here the piston of the compressor engaged for the simulation moving with frequency 60 Hz.

2.1.4 Dimension – 1 with frequency 100 Hz:

Here the piston of the compressor engaged for the simulation moving with frequency 100 Hz.

2.2 FREQUENCY STUDY WITH DIMENSION – 2:

- Different dimensions taken for the simulation are as follows,

Components	Radius (m)	Length (m)
Compressor (A)	0.00954	0.0075
Transfer line (B)	0.00155	0.101
After Cooler (C)	0.004	0.02
Regenerator (D)	0.004	0.058
Cold-heat exchanger (E)	0.003	0.0057
Pulse-tube (F)	0.0025	0.06
Warm-heat-exchanger (G)	0.004	0.01
Inertance-tube (H)	0.000425	0.684
Surge volume (I)	0.013	0.13

(Table 1.2)

2.2.1 Dimension – 2 with frequency 34 Hz:

Here the piston of the compressor engaged for the simulation moving with frequency 34 Hz.

2.2.2 Dimension – 2 with frequency 20 Hz:

Here the piston of the compressor engaged for the simulation moving with frequency 20 Hz.

Chapter – 3

METHODOLOGY

This chapter deals with the methodology adopted to solve the presently defined problems. The inertance tube pulse tube refrigerators (ITPTR) functioning under specified boundary conditions are modelled using Finite Volume Method (FVM). Dynamic Meshing and user define function (UDF) are used to modelled the Piston movement within the compressor. 2D axis-symmetric model has been used for the numerical simulation of ITPTR. Two-dimensional view of ITPTR, Two-dimensional view of axis-symmetry geometry of ITPTR and Enlarged view of the axis-symmetric meshed geometry are shown in Figures 2.1, 2.2 & 2.3 respectively,

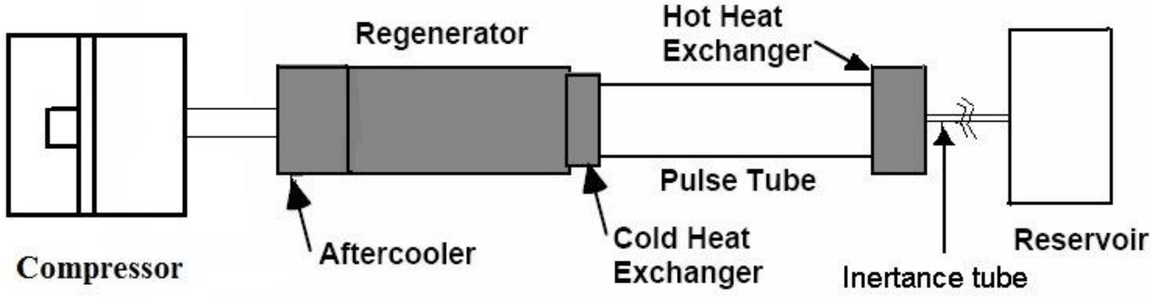


Figure 3.1: Two-dimensional view of ITPTR.

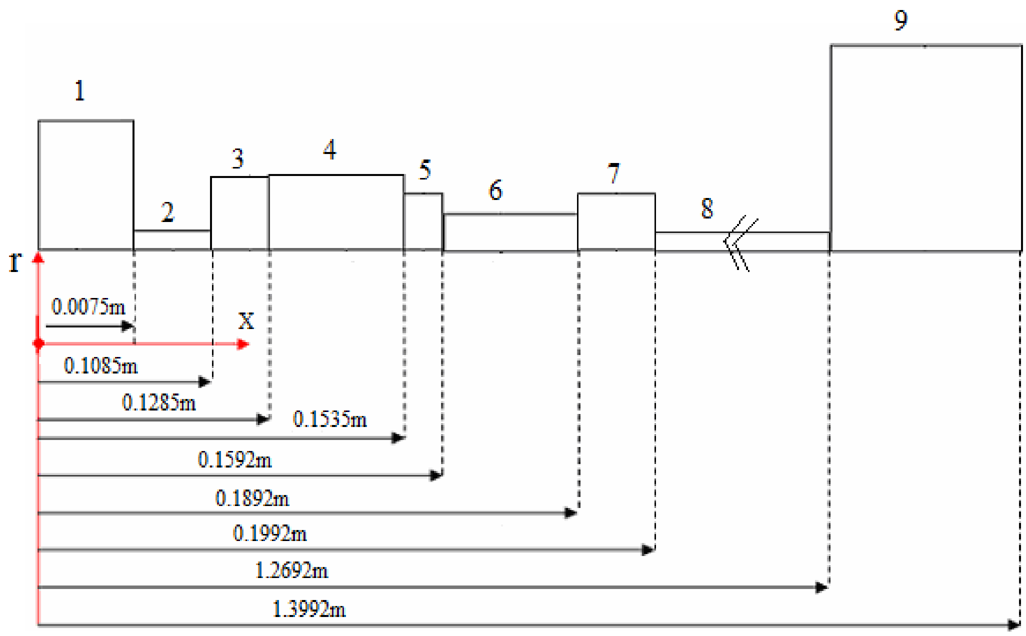


Figure 3.2: Two-dimensional view of axis-symmetry geometry of ITPTR.

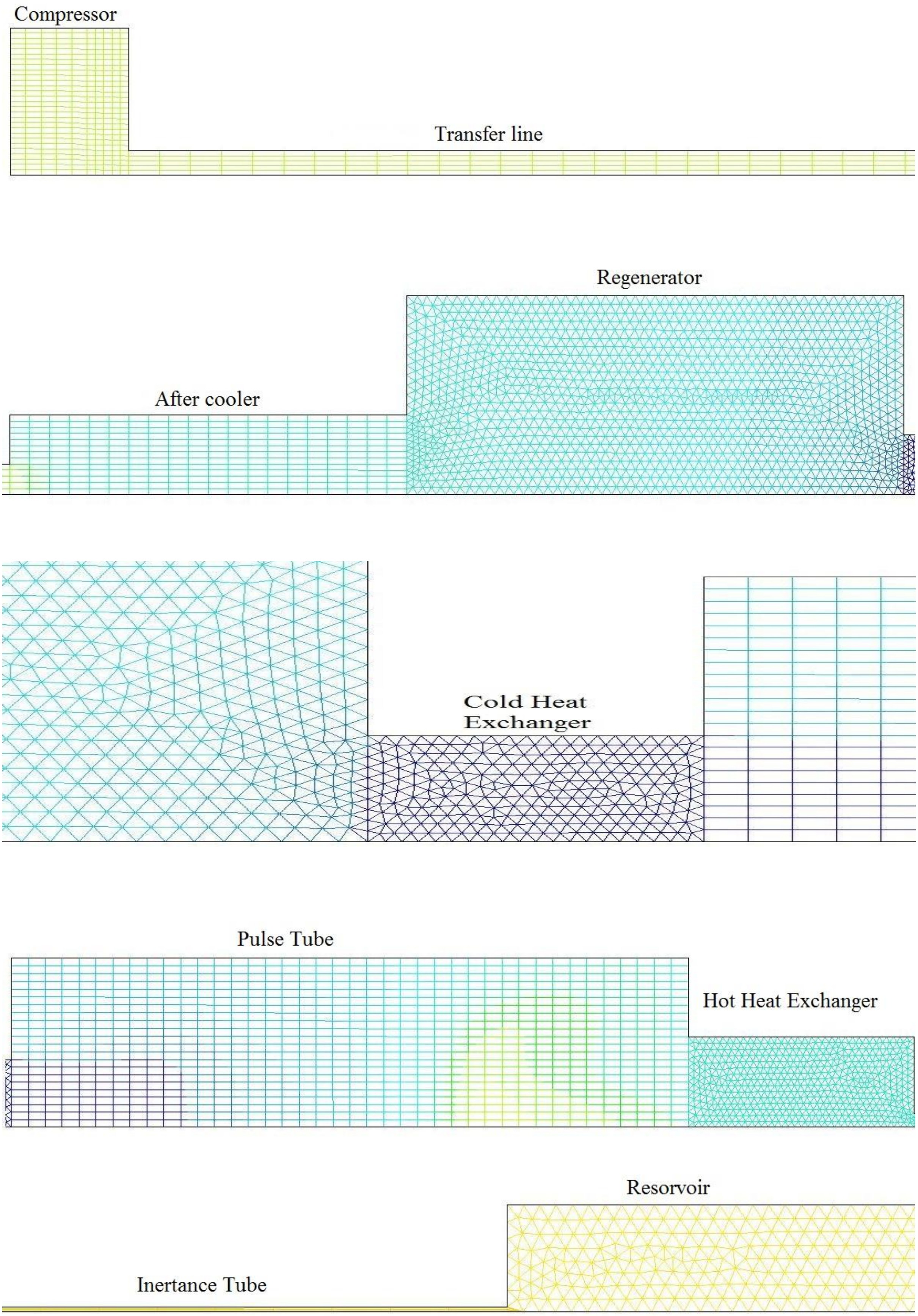


Figure 3.3: Enlarged view of the axis-symmetric meshed geometry.

3.1 FINITE VOLUME METHOD (FVM):

The Finite Volume Method (FVM) is the most useful and unique techniques used for explaining governing equations for the flow of fluid and heat and mass transfer problems. The most exciting features of the FVM are that the subsequent explanation satisfies the parameters such as energy, mass, momentum etc. A coarse grid result shows the exact integral balances. Apart from this, it can be applied to any type of grids (structured or unstructured, Cartesian, coarse, fine) and especially complex geometries. Hence, it is the policy for most of the commercial packages like Ansys, CHX and CFX etc. which are used to solve the flow of fluid and heat and mass transfer problems. In the finite volume method, the solution domain is subdivided into continuous compartments or control volumes where the variable of interests is situated at the centroid of the control volume forming a grid. The next phase is to integrate the differential form of the governing equations over each control volume. There are several patterns that can be used for interpolation, e.g. central differencing, upwind differencing, power-law differencing and quadratic upwind differencing schemes. The equations are called the discretized equation. In this manner the discretized equation states the conservation opinion for the variable within the control volume. These variables form a set of algebraic calculations which are resolved simultaneously by means of special algorithm.

3.1.1 The specified boundary conditions & for different parts of PTR:

Components	Material Selection	Boundary Conditions
Compressor	Steel	Adiabatic
Transfer line	Steel	Adiabatic
After Cooler	copper	T=293K
Regenerator	Steel	Adiabatic
Cold heat-exchanger	copper	Adiabatic
Pulse-tube	Steel	Adiabatic
Warm-heat-exchanger	copper	T=293K
Inertance-tube	Steel	Adiabatic
Surge volume	Steel	Adiabatic

(Table 3.1)

3.1.2 Initial conditions provided for the simulation:

Viscous-resistance(m^{-2})	94433962264 or 9.44e+9
Inertial-resistance(m^{-1})	76090
Initial Pressure (Pa)	3000000
Initial Temperature(K)	300
cold end load (W)	0
Porosity	0.69

(Table 3.2)

3.1.3 Governing Equations for PTR:

Different governing equations related to pulse tube are described below,

3.1.3.1 Conservation of mass equation:

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho\vec{v}) = S_m \quad (1.1)$$

Where

∇= Gradient operator

ρ= Density of the gas

$\vec{v}$ = Velocity in vector form

S_m = Source term

t = Time

(S_m = 0 for this case)

$$\frac{\partial \rho}{\partial t} + \frac{1}{r}\frac{\partial}{\partial r}(r\rho v_r) + \frac{\partial}{\partial x}(\rho v_x) = 0 \qquad (1.2)$$

Where

r = Radial coordinate

x = Axial coordinate

v_r = Velocity in radial coordinate

v_x= Velocity in axial coordinate

3.1.3.2 Conservation of Momentum:

$$\frac{\partial}{\partial t}(\rho\vec{v}) + \nabla(\rho\vec{v}\vec{v}) = -\nabla p + \nabla(\tau) + \rho\vec{g} + \vec{F} \qquad (1.3)$$

Where:

p = Static pressure

τ= Stress Tensors

$\vec{g}$= Acceleration due to gravity

$\vec{F}$= Source terms e.g. terms associated with porous media.

Assuming the fluid flow inside the PTR is Newtonian and the relation between shear stress and shear strain rate is:

$$\tau = \mu\,[(\nabla\vec{v} + \nabla\vec{v}^T) - \frac{2}{3}\nabla.\vec{v}I \qquad (1.4)$$

Where:

τ = Fluid molecular viscosity

I = Unit (Identity) tensor

T = Transpose

The momentum equations in axial & radial directions are as follows,

$$\frac{\partial}{\partial t}(\rho v_x) + \frac{1}{r}\frac{\partial}{\partial x}(r\rho v_x v_x) + \frac{1}{r}\frac{\partial}{\partial r}(r\rho v_r v_x)$$
$$= -\frac{\partial p}{\partial x} + \frac{1}{r}\frac{\partial}{\partial x}\left[r\mu\left(2\frac{dv_x}{dx} - \frac{2}{3}(\nabla.\vec{v})\right)\right]$$
$$+ \frac{1}{r}\frac{\partial}{\partial r}\left[r\mu\left(\frac{dv_x}{dr} + \frac{dv_r}{dx}\right)\right] \tag{2.1}$$

$$\frac{\partial}{\partial t}(\rho v_r) + \frac{1}{r}\frac{\partial}{\partial x}(r\rho v_x v_r) + \frac{1}{r}\frac{\partial}{\partial r}(r\rho v_r v_r)$$
$$= -\frac{\partial p}{\partial r} + \frac{1}{r}\frac{\partial}{\partial r}\left[r\mu\left(2\frac{dv_r}{dr} - \frac{2}{3}(\nabla.\vec{v})\right)\right] + \frac{1}{r}\frac{\partial}{\partial x}\left[r\mu\left(\frac{dv_r}{dx} + \frac{dv_x}{dr}\right)\right] - 2\mu\frac{v_r}{r^2}$$
$$+ \frac{2}{3}\frac{\mu}{r}(\nabla.\vec{v}) \tag{2.2}$$

Where v_r and v_x are the of radial and axial velocity vector componentsrespectively.

Where $\nabla.\vec{v} = \frac{\partial v_x}{\partial x} + \frac{\partial v_r}{\partial r} + \frac{v_r}{r}$ (2.3)

3.1.3.3 Conservation of energy:

$$\frac{\partial}{\partial t}(\rho E) + \nabla\{\vec{v}(\rho E + P)\} = \nabla.\left\{K_{eff}\nabla T - \sum_j h_j\vec{\Psi}_j + (\tau.\vec{v})\right\} + S \tag{2.4}$$

$$E = h - \frac{p}{\rho} + \frac{v^2}{2}$$

$$h = \int_{T_1}^{T} C_p\, dT$$

$k_{eff} = k + k_t$

k = Gas thermal conductivity

Where,

k_t= Turbulence thermal conductivity

C_P = Specific heat of gas

h = Local enthalpy

T = Temperature of the gas

V = Local velocity

$\overrightarrow{\Psi_J}$= Diffusion flux of species

S = Source term which can be caused by chemical reactions or volumetric heat generation.

For ideal gas behaviour,

$$P = \rho RT \tag{2.5}$$

Where R = Gas constant

3.1.4 Porous Media:

The different parts of PTR such as after cooler, regenerator, cold heat-exchanger and hot heat-exchanger can be modelled using porous media methods. The phenomenon is modelled using the volume-averaged conservation equations for mass and momentum.

The volume-averaged equation for mass conservation is,

$$\frac{\partial}{\partial t}(\gamma\rho) + \nabla.(\gamma\rho\vec{v}) = 0 \tag{2.6}$$

γ – Porosity

The volume–averaged equation of momentum can be written in two ways as below,

$$F_{porous\,x} = -\left(\frac{\mu}{\alpha}v_x + \frac{1}{2}C\rho\,\left|\,\vec{v}\,\right|\,v_x\right) \tag{2.7}$$

$$F_{porous\,r} = -\left(\frac{\mu}{\alpha}v_x + \frac{1}{2}C\rho\,\left|\,\vec{v}\,\right|\,v_r\right) \tag{2.8}$$

Where

μ= Dynamic viscosity of fluid

α= Permeability

C = Inertial resistance factor

$\vec{v}$ = Velocity

Again the porous –medium momentum equation, in vector form, can be represented as,

$$\frac{\partial}{\partial t}(\gamma\rho\vec{v}) + \nabla.(\gamma\,\rho\vec{v}\,\vec{v}) = -\gamma\nabla_p + \nabla.(\gamma\tau) - \left(\frac{\mu}{\alpha}\vec{v} + \frac{1}{2}C\rho\,\left|\,\vec{v}\,\right|\,v\right) \tag{2.9}$$

Where $\vec{v}$ is the physical velocity related to superficial velocity $\overrightarrow{v_1}$ as,

$\overrightarrow{v_1} = \gamma\vec{v}$

➢ The axial and radial components of the above equation are,

3.1.4.1 Porous–medium momentum equation in axial direction:

$$\frac{\partial}{\partial t}(\gamma\rho v_x) + \frac{1}{r}\frac{\partial}{\partial x}(\gamma r\rho v_x v_x) + \frac{1}{r}\frac{\partial}{\partial r}(\gamma r\rho v_r v_x)$$

$$= -\frac{\partial}{\partial x}(\gamma p) + \frac{1}{r}\frac{\partial}{\partial x}\left[r\mu\left(2\frac{\partial(\gamma v_x)}{\partial x} - \frac{2}{3}\left(\nabla.(\gamma\vec{v})\right)\right)\right]$$

$$+\frac{1}{r}\frac{\partial}{\partial r}\left[r\mu\left(\frac{\partial(\gamma v_x)}{\partial r} + \frac{\partial(\gamma v_r)}{\partial x}\right)\right] - \left(\frac{\mu}{\alpha}v_x + \frac{1}{2}C\rho \mid \vec{v} \mid v_x\right) \qquad (2.10)$$

3.1.4.2 Porous–medium momentum equation in radial direction:

$$\frac{\partial}{\partial t}(\gamma\rho v_r) + \frac{1}{r}\frac{\partial}{\partial x}(\gamma r\rho v_x v_r) + \frac{1}{r}\frac{\partial}{\partial r}(\gamma r\rho v_r v_r)$$

$$= -\frac{\partial}{\partial r}(\gamma p) + \frac{1}{r}\frac{\partial}{\partial r}\left[r\mu\left(2\frac{\partial(\gamma v_r)}{\partial r} - \frac{2}{3}\left(\nabla.(\gamma\vec{v})\right)\right)\right] + \frac{1}{r}\frac{\partial}{\partial x}\left[r\mu\left(\frac{\partial v_r}{\partial x} + \frac{\partial v_x}{\partial r}\right)\right]$$

$$-2\mu\frac{v_r}{r^2} + \frac{2}{3}\frac{\mu}{r}(\nabla.\vec{v})$$

$$-\left(\frac{\mu}{\alpha}v_r + \frac{1}{2}C\rho \mid \vec{v} \mid v_r\right) \qquad (2.11)$$

Now a days heat transfer in porous media has been fascinating the attention of large number of investigators. The impact of porous media can be studied is two ways

- To improve the performance of existing porous-media-related thermal systems.
- To generate new ideas and explore new awareness with respect to the use of porous media in heat and fluid flow applications.

Porous media is defined as a material consisting of a solid matrix with interconnected voids (pores). The interconnected channels allow passage of one or more fluids. Examples of flow of fluid and heat transfer occur in a porous media are: Underground water movement in soils, Filtration of gaseous reactance in a catalyst pore, cryo-cooler regenerators etc.

Porosity measures the empty spaces in a material. It is the ratio between volumes of voids upon total volume. The porosity of any material should be within zero to one. Thermal reporting in porous region,

$$\frac{\partial}{\partial t}\left(\gamma\rho_f E_f + (1-\gamma)\rho_s E_s\right) + \nabla.\left(\vec{v}\left(\rho_f E_f + P\right)\right) = \nabla.\left(k_{eff}\nabla T + \tau\vec{v}\right) \qquad (2.12)$$

Where,

$k_{eff} = \gamma K_S + (1-\gamma)K_f$

k_{eff} – Effective thermal conductivity

γ – Porosity

K_S – Thermal conductivity in solid phase

K_f - Thermal conductivity in solid phase

3.1.5 Figure of Merit (FOM):

FOM is a factor which uses to calculate the performance of PTR. For an ideal gas the figure of merit for pulse tube is defined as,

$$FOM = \left(H/P_d\right)V \qquad (2.13)$$

Where,

H - Enthalpy flow

P_d–Pressure indynamic condition

V – Volume flow-rate

FOM varies between 0.55 - 0.85 for small size pulse tubes whereas for large pulse tubes are about 0.96.

3.2 DYNAMIC MESHING FUNCTION:

Ansys 13.0 has a dynamic meshing role. The dynamic mesh function in Ansys used to model different flows where the shape of the model is varying with respect to time because of the motion on the shape boundaries like in reciprocating compressor when piston moves the domain of fluid will changes with time in case of compression and expansion of fluid. This type of model could be handled in Ansys by using dynamic meshing function. The update of the model meshing is handled automatically by Ansys at each time step created on the new locations of the boundaries. For using the model of dynamic meshing, it is needed to provide a starting volume mesh of the motion of any affecting zones in the model. Ansys permits describing the motion using either boundary profiles or user defined function (UDF).

In order to model the piston and cylinder, Ansys dynamic meshing function is used. A user defined function (UDF) is developed in C coding language to simulate the piston cylinder effect. The reciprocating compressor used in the above simulation. The piston head motion is accordingly found from the following equation,

Piston displacement is expressed as,

$$X = X_a \sin(\omega t) \tag{3.1}$$

Differentiating equation with respect to time,

$$\mathrm{V} = \frac{\mathrm{dx}}{\mathrm{dt}} = X_a \omega \sin \omega t \tag{3.2}$$

Where,

X - Piston displacement

X_a - Piston displacement amplitude

ω - Angular Frequency in rad/sec (=2πf)

F – Frequency in Hertz

t – Time period in second

3.3 USER DEFINED FUNCTION (UDF):

The dynamic mesh options of the Ansys are used to model the compressor of Stirling type ITPTR. The user defined function (UDF) could be written by using visual studio package. The UDF is saved by giving suitable name like **“velocity.c”.** This is stored in the same folder where mesh files are saved. The UDF needs to be compiled and then linked with piston which makes the reciprocating motion possible on the piston.

The velocity UDF for piston head motion is as follows:

```
#include "udf.h"

DEFINE_CG_MOTION (vel_comp, dt, vel, omega, time, dtime)

{
    Real freq = 34.0;
    real w = 2.0 * M_PI * freq;
    real Xcomp = 0.004511;
    /*reset velocities*/
    NV_S(vel, = ,0.0);
    NV_S(omega, = ,0.0);
    vel[0] = w * Xcomp * cos (w * time);
}
```

- After compiling the UDF by using the Ansys package the mesh motion preview checked for the piston by providing different time step size then choose a suitable time step size for the dynamic meshing. After that we can able to start our simulation by specifying a value for number of iteration.

Chapter–4

Results & Discussion

This chapter contains the results obtained for different cases as described in section-2.

Numerical investigation of single stage Stirling type inertance tube pulse tube refrigerator (ITPTR) is performed by using commercial Ansys package by means of a CFD code. On the basis of simulation performance of two models are compared. In first model there are four different cases having same boundary condition but having different frequency are studied whereas in second model there are two different cases having same boundary conditions and different frequencies are studied. CFD simulation results are discussed in this section by considering the surface temperature of cold heat exchanger with no load condition. Various graphs are plotted for each frequency are shown below such as cooling behaviour of cold heat-exchanger, cold heat-exchanger surface temperature variation in steady state condition, contour diagram for temperature & density, Temperature distribution for cold heat-exchanger.

4.1 VALIDATION:

Firstly the validation takes place by using the simulation model observed in cha et al. (2006) with present work for 83 Seconds which is plotted below. By comparing the cooling behaviour of the above two cases it is found that they are perfectly matched.

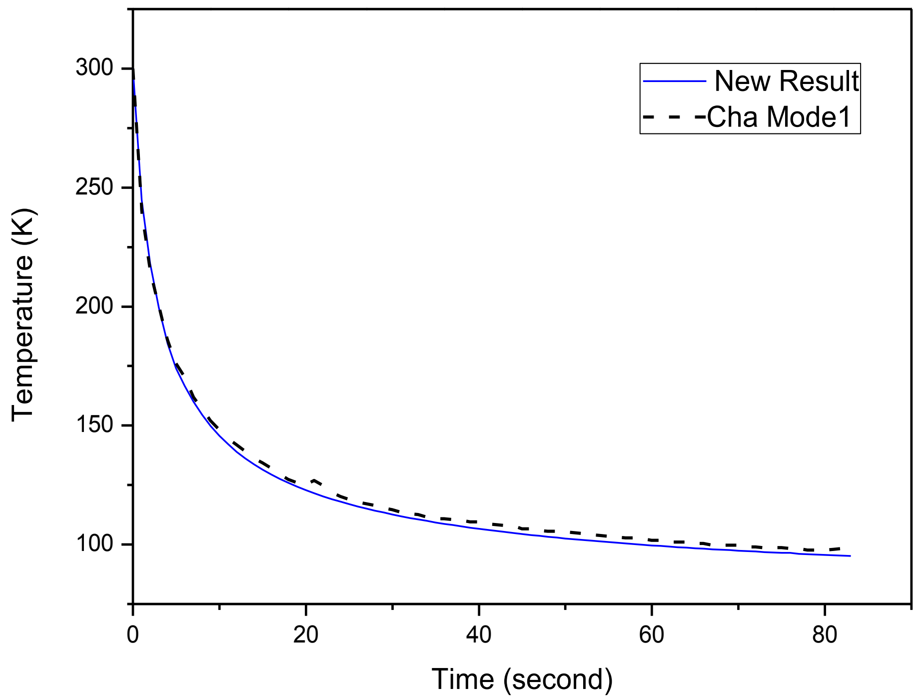


Figure 4.1: Comparison between Cha et al. (2006) & present work

4.2 FREQUENCY STUDY WITH DIMENSION–1:

The variations of cyclic average temperature of cold heat exchanger with respect to time for different frequencies of compressor are shown below for dimension-1 described in section-2. Each case corresponds to adiabatic boundary condition at cold end heat exchanger with no load condition. All the simulations are carried out by assuming an initial pressure of 30 bars & initial temperature of 300K for 60 seconds till the cooling behaviour of cold heat exchanger reached to steady state condition. Each case consist of various curves such as cooling behaviour of cold heat exchanger, cold heat-exchanger surface temperature variation in steady state condition, contour diagram for temperature, contour diagram for density, temperature distribution for cold heat exchanger.

4.2.1 ITPTR with Dimension–1 with frequency 10 Hz:

This case characterises the simulation of ITPTR maintaining the cold heat exchanger at no load condition. Here piston of the compressor moving with frequency 10Hz. The Cooling behaviour of the cold end heat exchanger for frequency of 10 Hz is shown below. Here temperature of cold end heat exchanger is 277.6259 K.

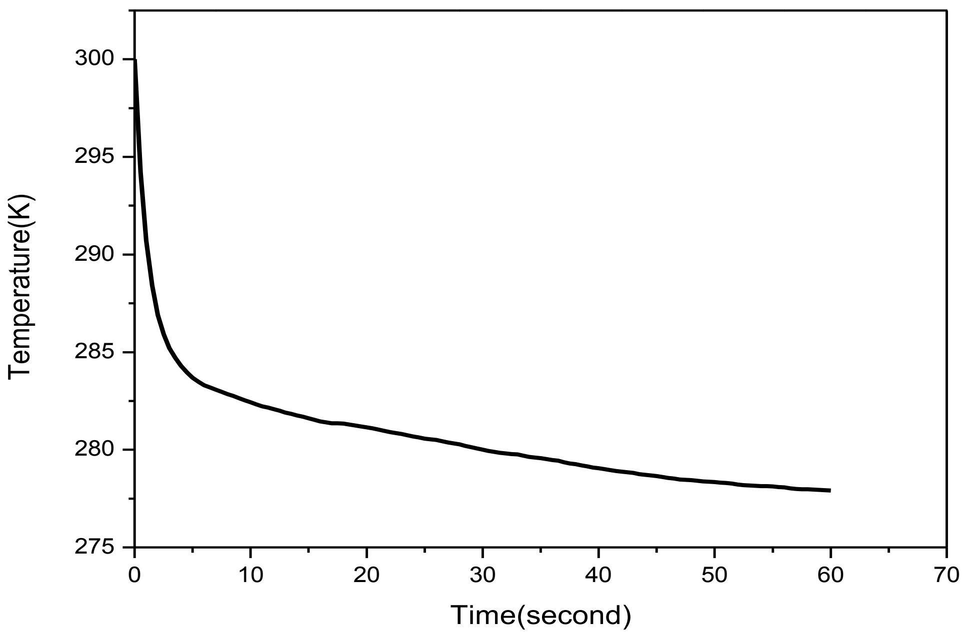


Figure 4.2: Cooling Behaviour of cold heat exchanger

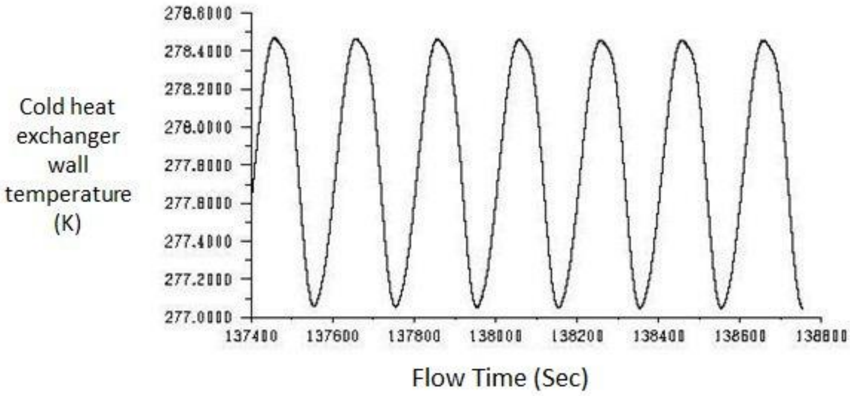


Figure 4.3: Cold heat-exchanger surface temperature variation in steady state condition

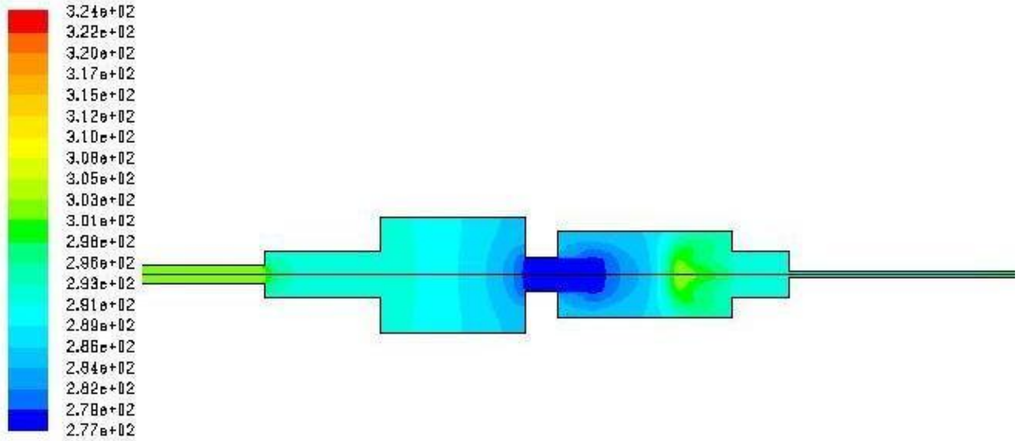


Figure 4.4: Contour diagram for temperature

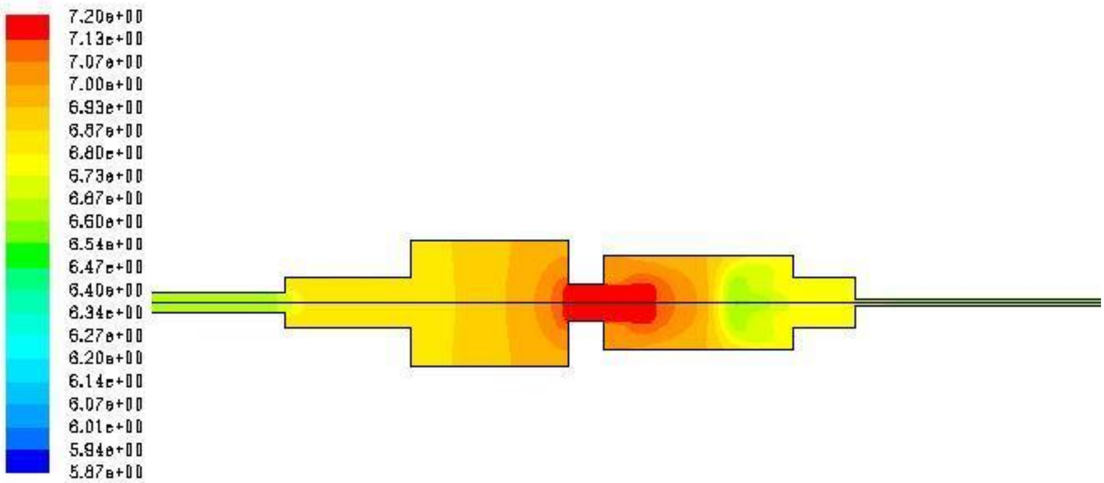


Figure 4.5: Contour diagram for density

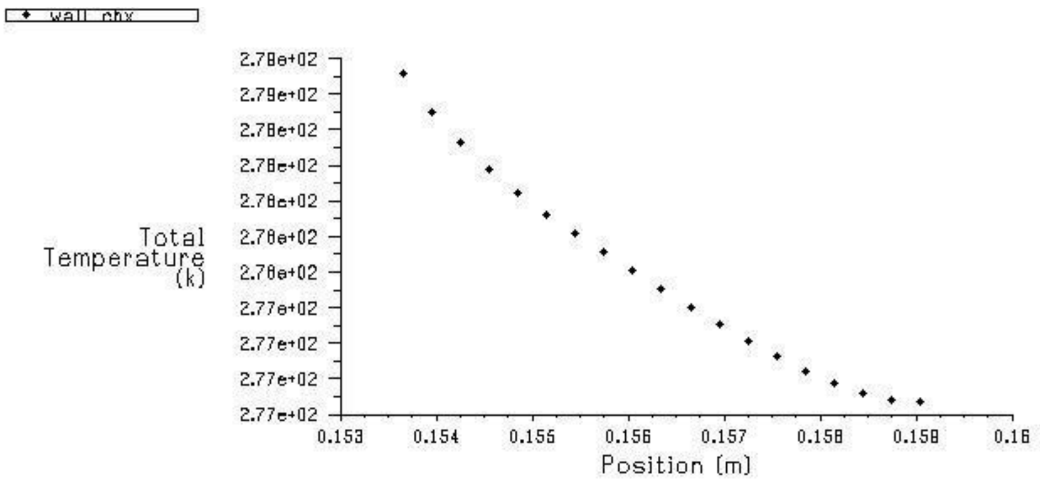


Figure 4.6: Temperature distribution for cold heat exchanger

4.2.2 ITPTR with Dimension–1 with frequency 20 Hz:

This case characterises the simulation of ITPTR maintaining the cold heat exchanger at no load condition. Here piston of the compressor moving with frequency 20 Hz. The Cooling behaviour of the cold end heat exchanger for frequency of 20 Hz is shown below. Here temperature of cold end heat exchanger is 264.9179 K.

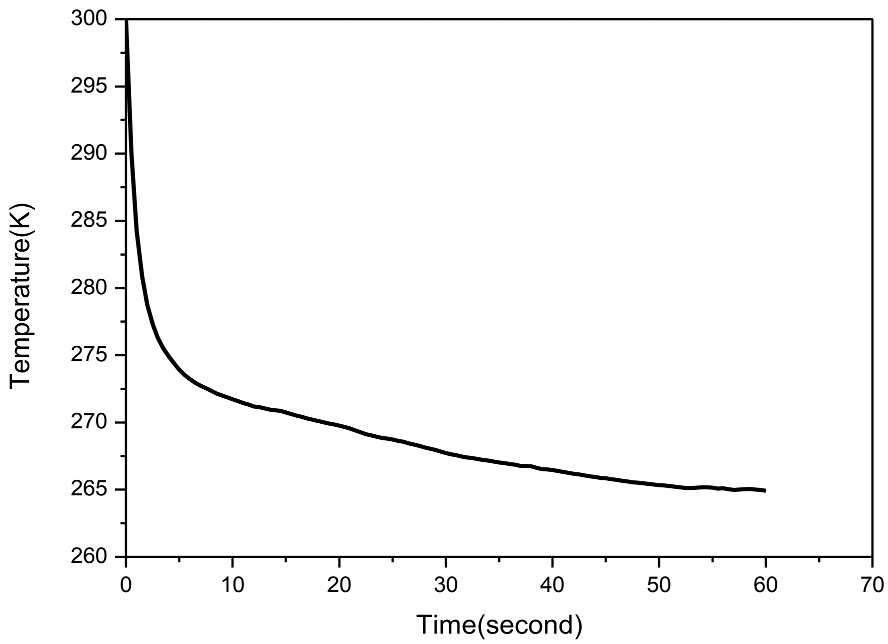


Figure 4.7: Cooling Behaviour of cold heat exchanger

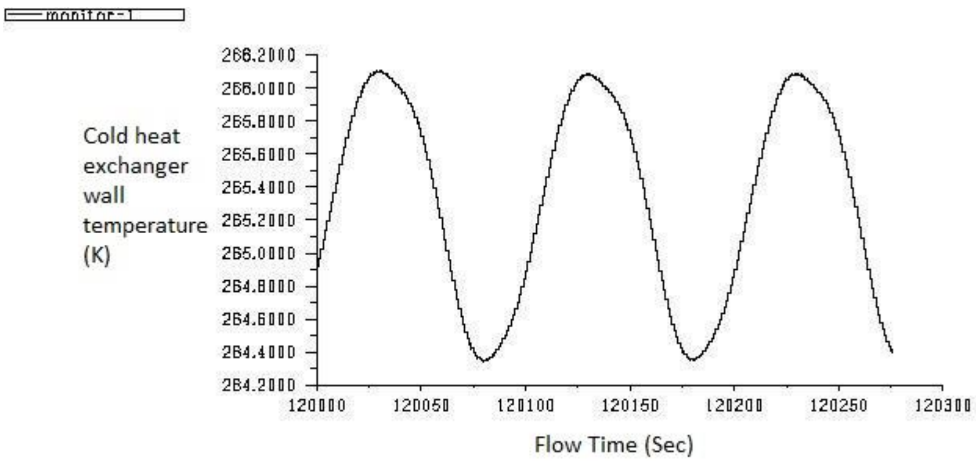


Figure 4.8: Cold heat-exchanger surface temperature variation in steady state condition

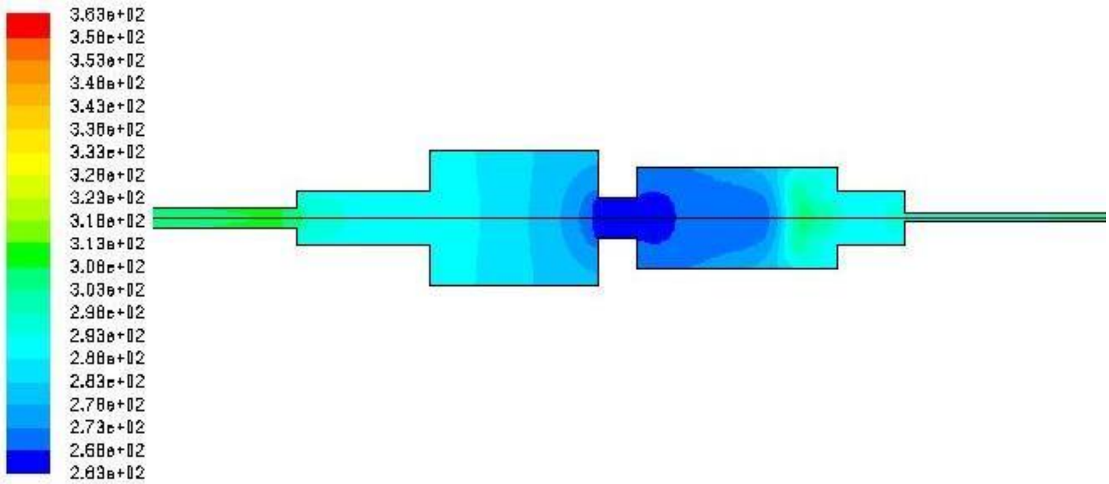


Figure 4.9: Temperature contour along axial direction

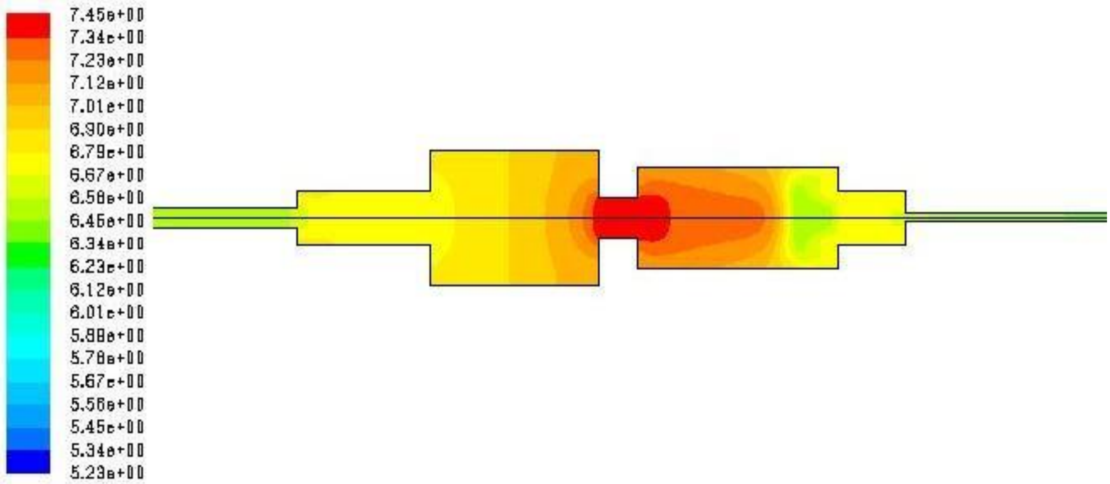


Figure 4.10: Density contour along axial direction

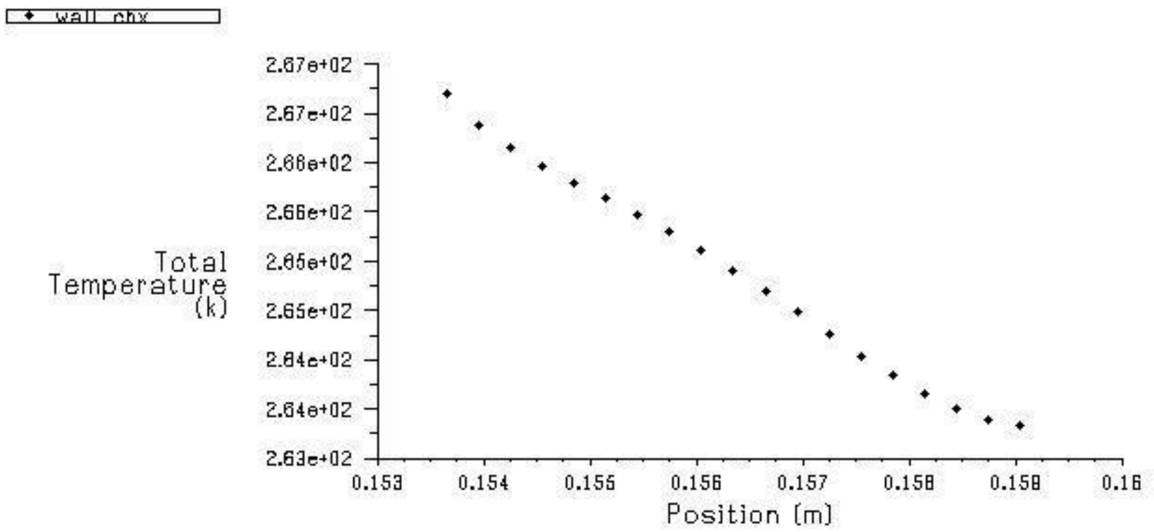


Figure 4.11: Temperature distribution for cold heat exchanger

4.2.3 ITPTR with Dimension–1 with frequency 60 Hz:

This case characterises the simulation of ITPTR maintaining the cold heat exchanger at no load condition. Here piston of the compressor moving with frequency 60 Hz. Here the piston frequency. The Cooling behaviour of the cold end heat exchanger for frequency of 60 Hz is shown below. Here temperature of cold end heat exchanger is 270.4995 K.

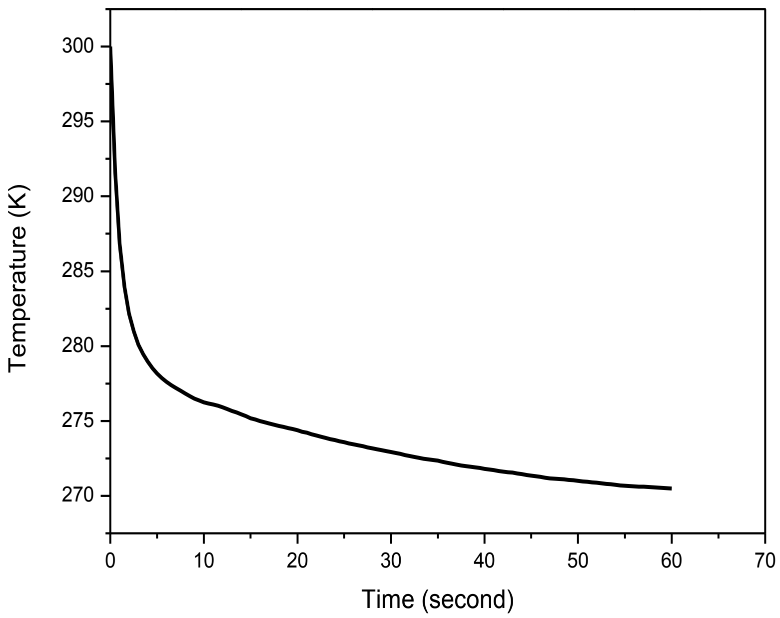


Figure 4.12: Cooling Behaviour of cold heat exchanger

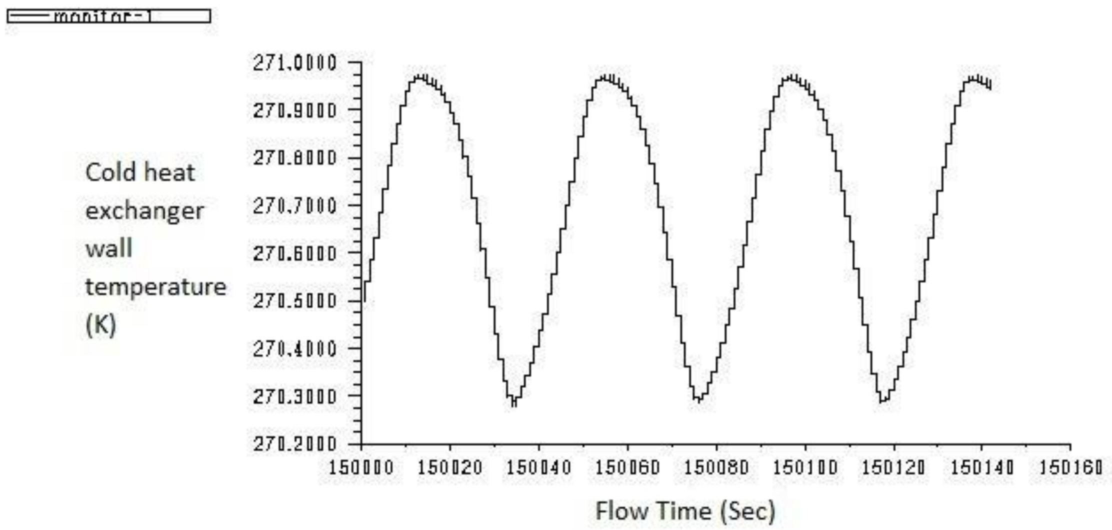


Figure 4.13: Cold heat-exchanger surface temperature variation in steady state condition

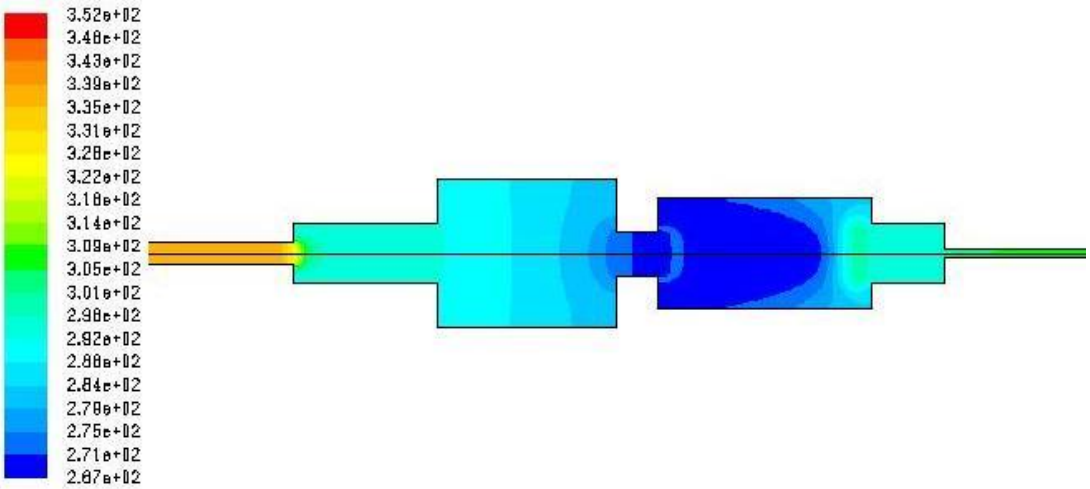


Figure 4.14: Temperature contour along axial direction

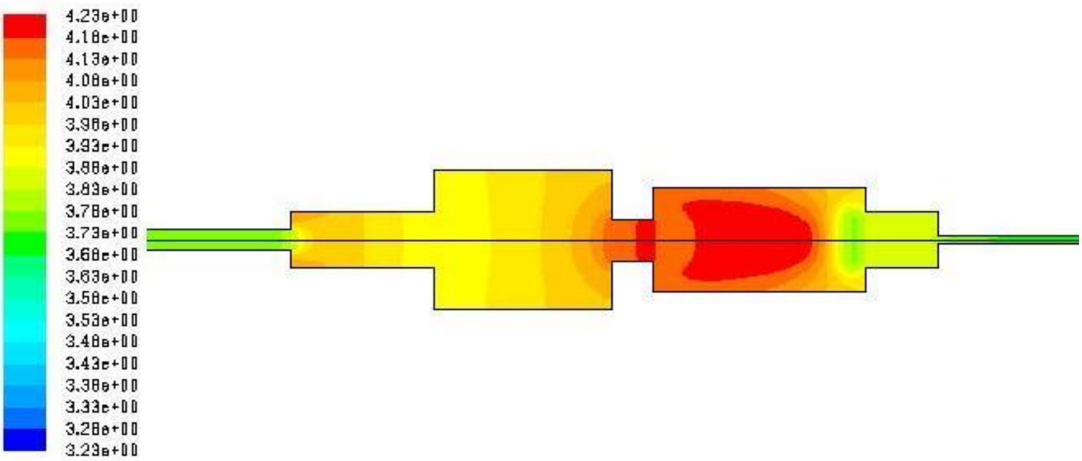


Figure 4.15: Density contour along axial direction

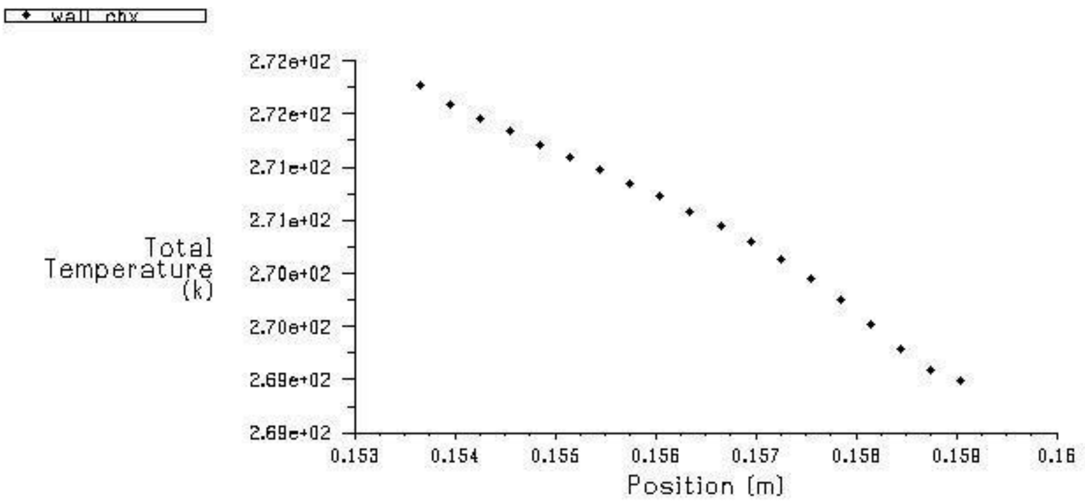


Figure 4.16: Temperature distribution for cold heat exchanger

4.2.4 ITPTR with Dimension–1 with frequency 100 Hz:

This case characterises the simulation of ITPTR maintaining the cold heat exchanger at no load condition. Here piston of the compressor moving with frequency 100 Hz. The Cooling behaviour of the cold end heat exchanger for frequency of 100 Hz is shown below. Here temperature of cold end heat exchanger is 274.2181 K.

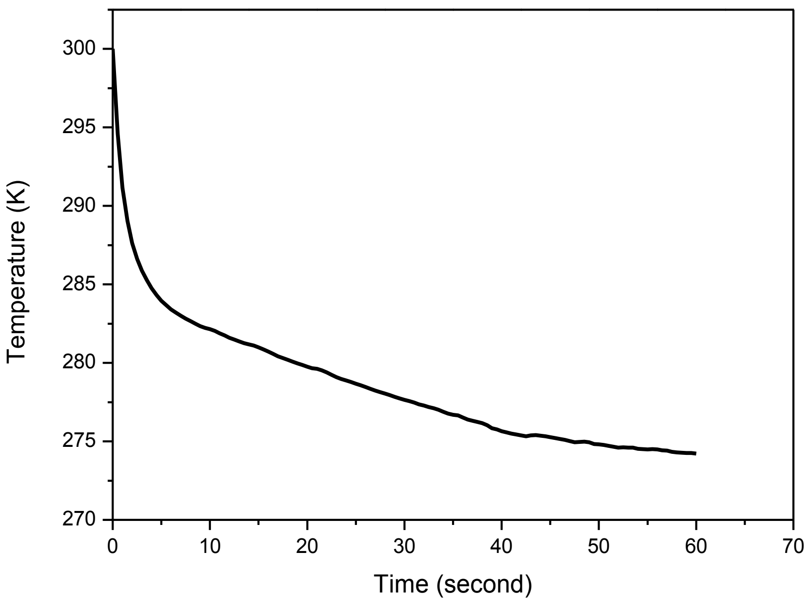


Figure 4.17: Cooling Behaviour of cold heat exchanger

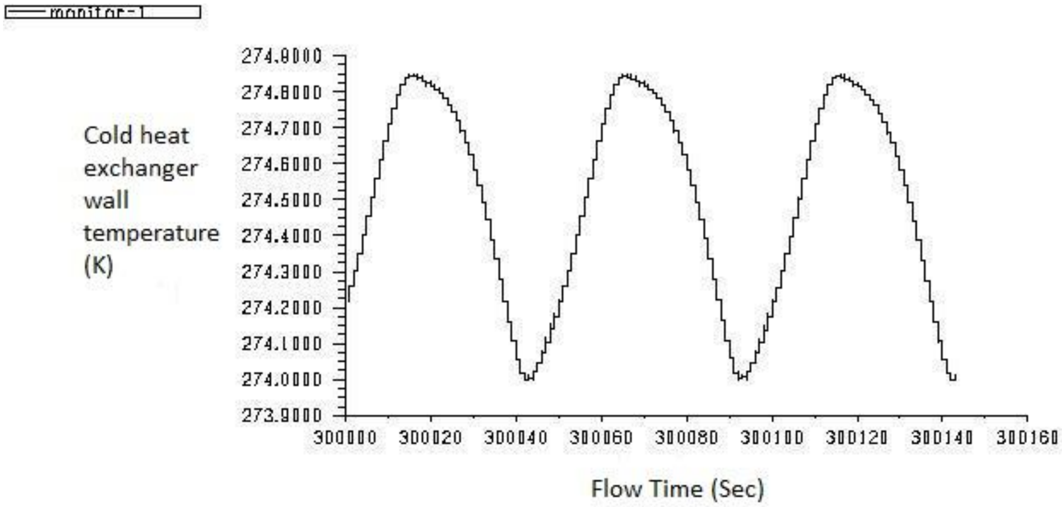


Figure 4.18: Cold heat-exchanger surface temperature variation in steady state condition

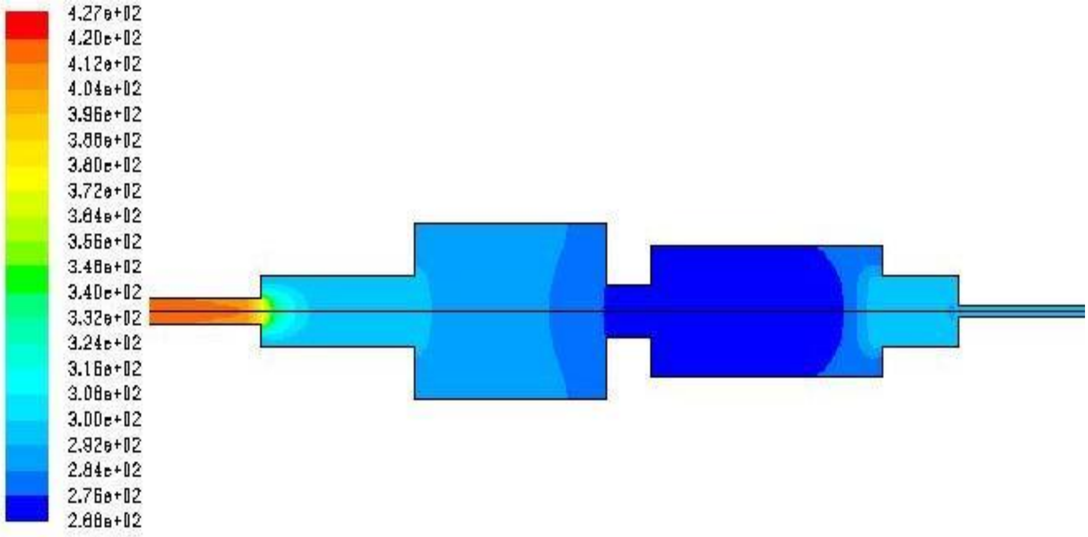


Figure 4.19: Temperature contour along axial direction

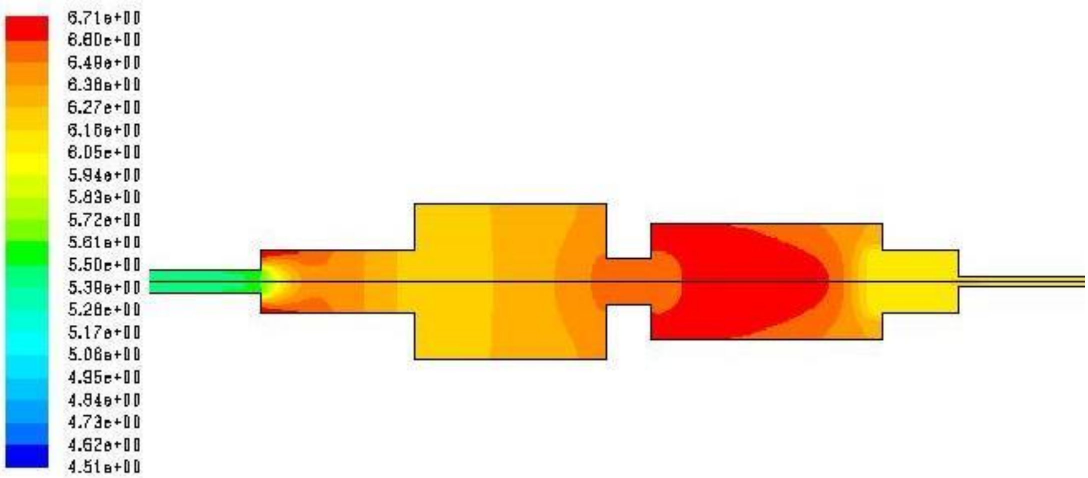


Figure 4.20: Density contour along axial direction

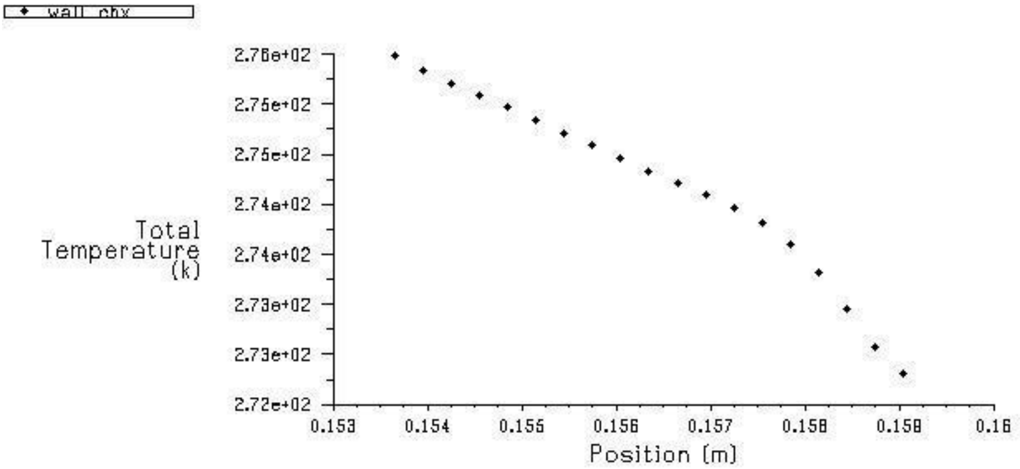


Figure 4.21: Temperature distribution for cold heat exchanger

4.2.5 Comparison for different frequency with Dimension–1:

From the above four frequency studies comparison among the cooling effect of all of them will be as follows. Here comparison carried out by plotting a graph between Time (in second) as abscissa (X-axis) & Temperature (in Kelvin) as ordinate (Y-axis). It is found that for the above dimension cooling rate for 20 Hz frequency is good comparatively others.

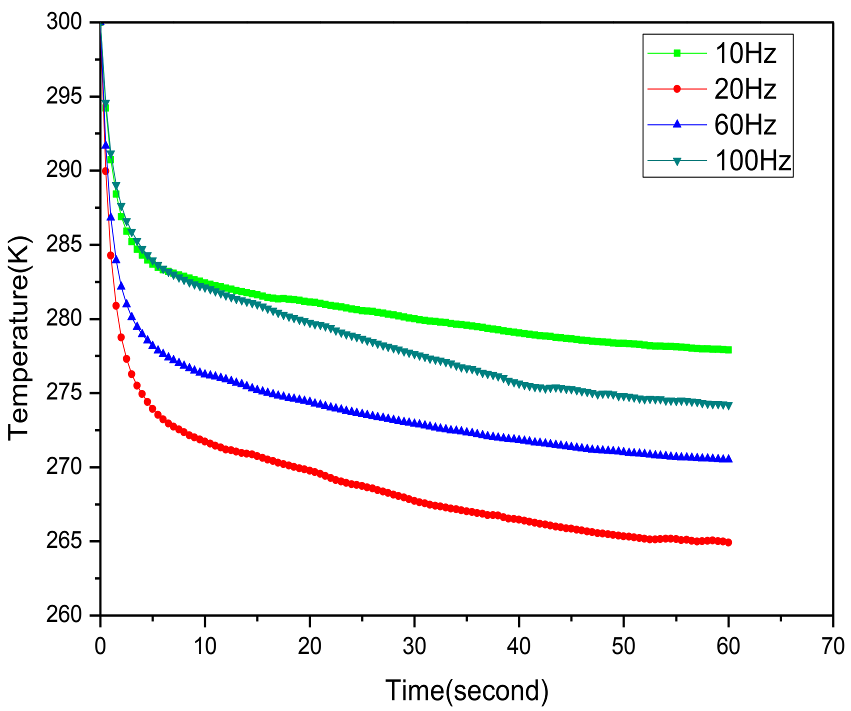


Figure 4.22: Comparison between cooling effects of all the frequencies for dimension-1

4.3 FREQUENCY STUDY WITH DIMENSION–2:

The variations of cyclic average temperature of cold heat exchanger with respect to time for different frequencies of compressor are shown below for dimension-1 described in section-2. Each case corresponds to adiabatic boundary condition at cold end heat exchanger with no load condition. All the simulations are carried out by assuming an initial pressure of 30 bars & initial temperature of 300 K. Each case consist of various curves such as cooling behaviour of cold heat exchanger, contour diagram for temperature, contour diagram for density, temperature distribution for cold heat exchanger.

4.3.1 ITPTR with Dimension – 2 having frequency 34 Hz:

Here piston of the compressor moving with frequency 34 Hz. The Cooling behaviour of the cold end heat exchanger for frequency of 100 Hz is shown below. Here temperature of cold end heat exchanger reached to 95.19126 K in 83 second.

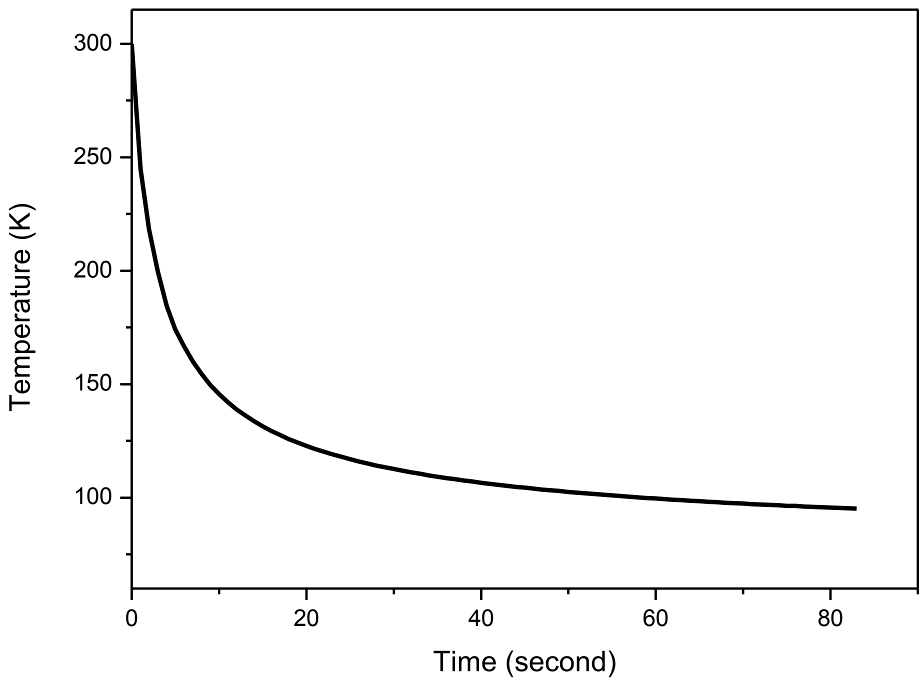


Figure 4.23: Cooling Behaviour of cold heat exchanger

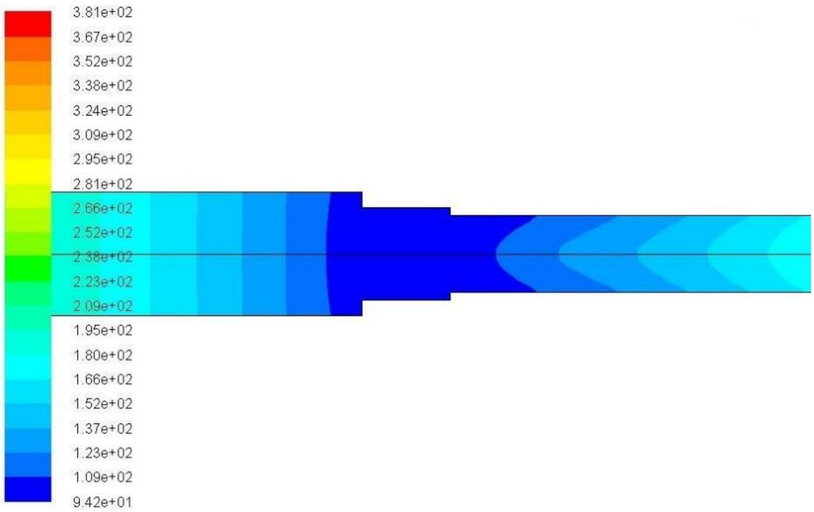


Figure 4.24: Temperature contour along axial direction for CHX

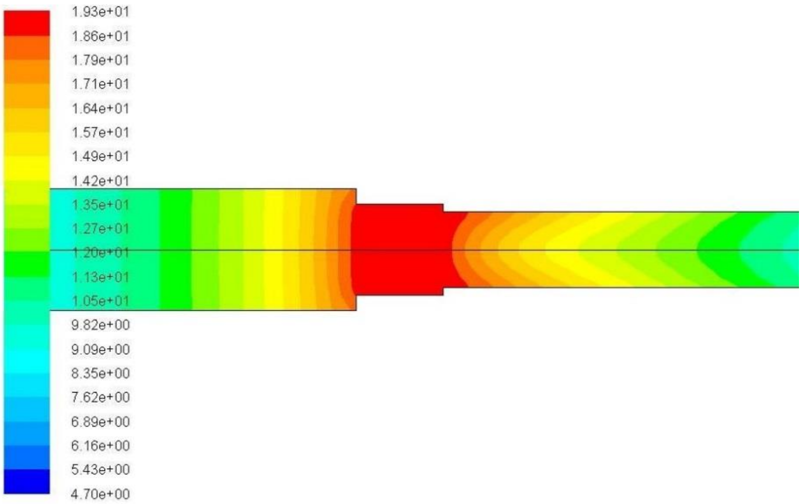


Figure 4.25: Density contour along axial direction for CHX

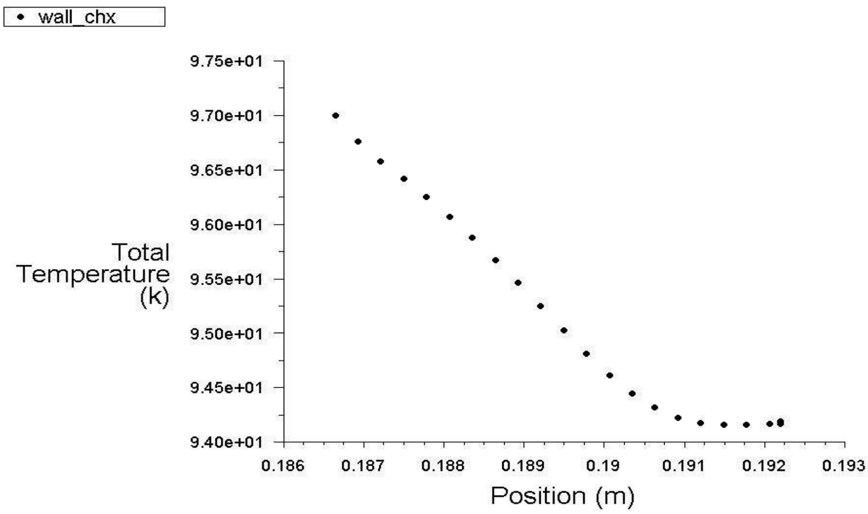


Figure 4.26: Temperature distribution for cold heat exchanger

4.3.2 ITPTR with Dimension–2with frequency 20 Hz:

Here piston of the compressor moving with frequency 34 Hz. The Cooling behaviour of the cold end heat exchanger for frequency of 100 Hz is shown below. Here cold end heat exchanger reached to130.2757 K in 36 second.

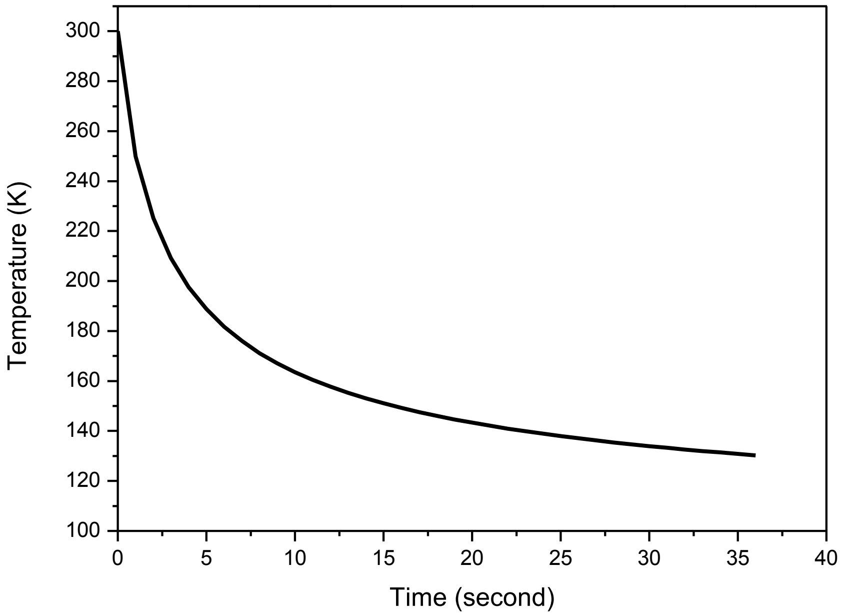


Figure 4.27: Cooling Behaviour of cold heat exchanger

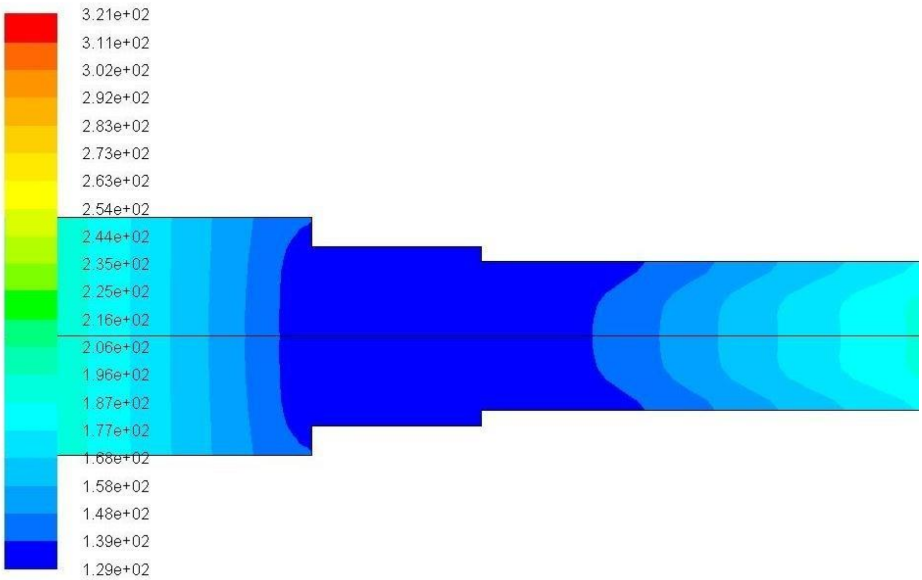


Figure 4.28: Temperature contour along axial direction for CHX

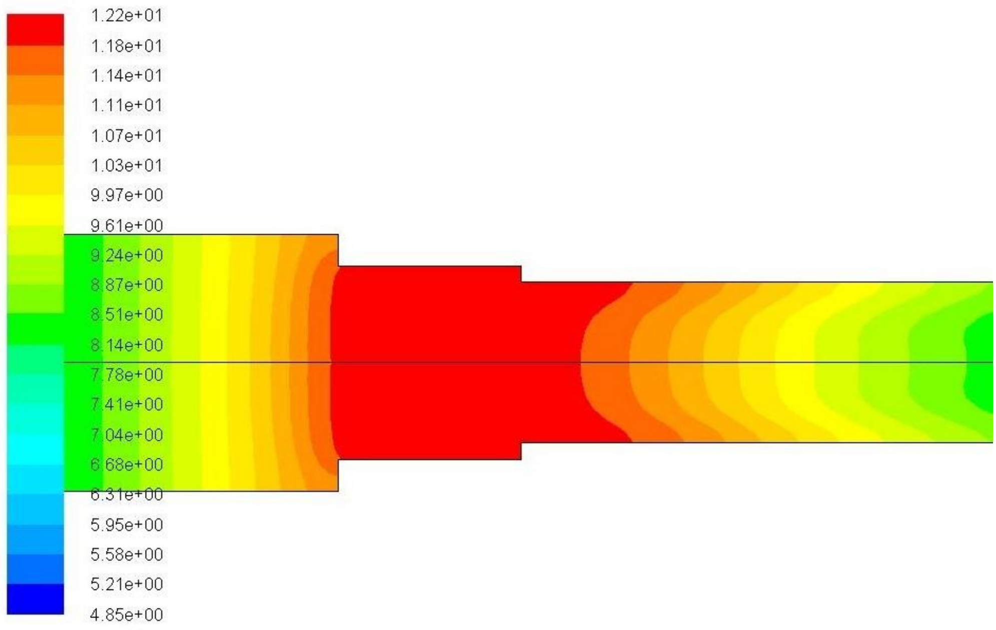


Figure 4.29: Density contour along axial direction

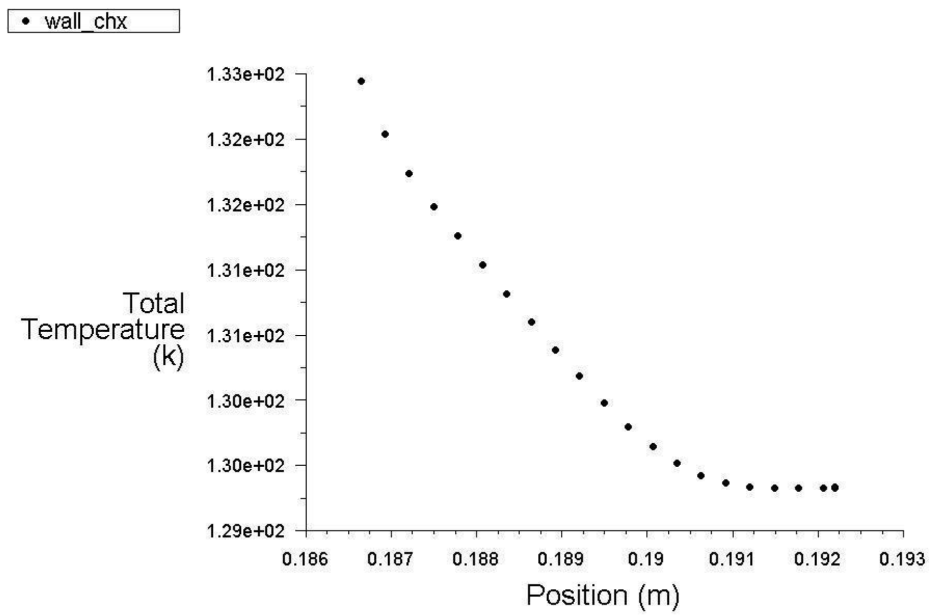


Figure 4.30: Temperature distribution for cold heat exchanger

4.2.3 Comparison for different frequency with Dimension – 2:

From the above two frequency studies, comparison between their cooling effects will be as follows. Here comparison carried out by plotting a graph between Time (in second) as abscissa (X-axis) & Temperature (in Kelvin) as ordinate (Y-axis). It is found that for the above dimension cooling rate for 34 Hz frequency is good comparatively others.

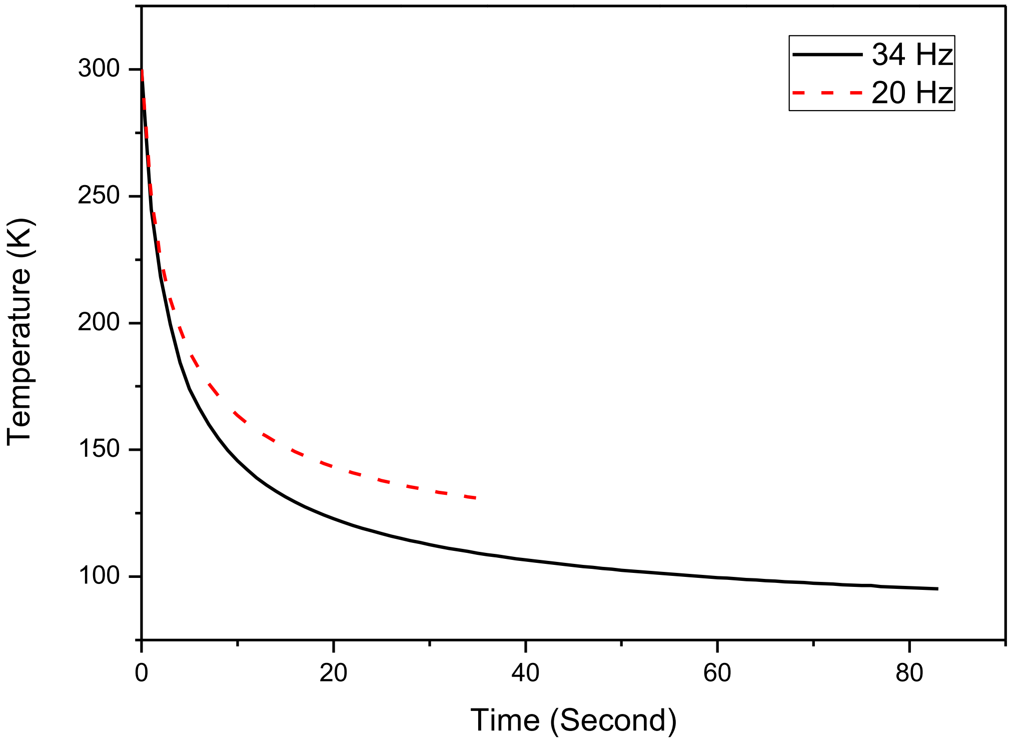


Figure 4.31: Comparison between cooling effects of all the frequencies for dimension-1

Chapter - 5

CONCLUSION & FUTURE SCOPE

5.1 CONCLUSION:

The numerical simulation on inertance tube PTR are carried out with a specified boundary conditions with varying the compressor frequency in different ways carried out by Finite Volume Method (FVM) using ANSYS package. It is found that the performance of the ITPTR and the cooling rate is strongly dependent on the frequency of the compressor.

At first the present numerical simulation validated with the model provided by Cha et al. (2006) with same compressor frequency (34Hz) and same boundary condition up to 83 seconds. By comparing the cooling behaviour of the above two cases it is found that they are perfectly matched.

Then the numerical study has been performed with a particular dimension of ITPTR by providing various compressor frequencies. Numerical study have been made for four different frequencies such as 10 Hz, 20 Hz, 60 Hz & 100 Hz for a particular dimension of ITPTR and it is found that the cooling behaviour at the surface of cold heat exchanger is quite good with 20 Hz frequency. All the studies are carried out for 60 seconds and the surface temperature of cold heat exchanger are found as 277.6259 K, 264.9179 K, 270.4995 K and 274.2181 K respectively.

Again the frequency study have been performed with a another dimension of ITPTR by providing two different compressor frequencies such as 20 Hz and 34 Hz. It is found that the cooling behaviour at the surface of cold heat exchanger is quite good with 34 Hz frequency. In 34 Hz frequency the surface temperature of cold heat exchanger are found as 95.19126 K in 83 seconds and in 20 Hz frequency the surface temperature of cold heat exchanger are found as130.2757 K in 36 seconds.

5.2 FUTURE SCOPE OF THE WORK:

Simulating the PTR by using different materials, different amplitude, different porosity, different boundary conditions and mixture working fluids.

Reference:

1. Gifford, W.E. and Longsworth, R.C. Pulse tube refrigeration, Trans ASME B J Eng Industry 86(1964), pp.264-267.
2. Mikulin, E.I., Tarasow, A.A. and Shkrebyonock, M.P. Low temperature expansion pulse tube, Advances in cryogenic engineering 29(1984), pp.629-637.
3. Zhou, Y. Zhu, W. and Sun, Y. Pulse tube with axial curvature, Advances in cryogenic engineering 33(1988), pp.860-868.
4. Radebaugh, R., Zimmermann, J.D., Smith, R. and Louie, B.A. Comparison of three types of pulse tube refrigerators: new methods for reaching 60K, Advances in cryogenic engineering 31(1986), pp.779-789.
5. Zhu Shaowei, Kakimi, Yasuhiro and Matsubara, Yoichi, Waiting time effect of a GM type orifice pulse tube refrigerator. Cryogenics, 38(1998), pp.619-624.
6. Wang, C., S. Wang, Q., Cai, J. H. and Yuan, Z., Experimental study of multi-bypass pulse-tube refrigerator, Cryogenics, 36(1996), pp. 555-558.
7. Huang, B.J. and Yu ,G. L., Experimental study on the design of orifice pulse tube refrigerator, International Journal of Refrigeration,24(2001),pp. 400-408.
8. Cha, J. S., et al. "Multi-dimensional flow effects in pulse tube refrigerators, "Cryogenics 46.9 (2006), pp. 658-665.
9. Wu, P.Y., Zhang, Li. Qian, L.L. and Zhang, L. Numerical modelling of orifice pulse tube refrigerator by using the method of characteristics, Advances in cryogenic engineering 39(1994), pp.1417-1431.
10. Xu, M. Y., de Waele, A. T. A. M. and Ju Y. L., A pulse tube refrigerator below 2K. Cryogenics, 39(1999), pp. 865-869.
11. Zhang X.B., L.M. Qiu, Z.H. Gan, Y.L. He, Effects of reservoir volume on performance of pulse tube cooler, International Journal of Refrigeration 30 (2007) pp.11-18.
12. Liang, J., Ravex, A. and Rolland, P., Study on pulse tube refrigeration Part 2:Theoretical modeling, Cryogenics, 36(1996), pp. 95-99.
13. Antao, Dion Savio, and Bakhtier Farouk. "Computational fluid dynamics simulations of an orifice type pulse tube refrigerator: Effects of operating frequency." *Cryogenics* (2011): 192-201.
14. Conrad, T. J., Seyed M. Ghiaasiaan, and Carl Scott Kirkconnell. "Simulation of Boundary Layer Effects in the Pulse Tube of a Miniature Cryo-cooler." (2008).

15. Farouk, Bakhtier, and Dion SavioAntao. "Numerical analysis of an OPTR: Optimization for space applications." *Cryogenics* 52.4 (2012): 196-204.
16. Wang, Chao, Peiyi Wu, and Zhongqi Chen. "Numerical modelling of an orifice pulse tube refrigerator." *Cryogenics* 32.9 (1992): 785-790.
17. Gan, Z. H., et al. "Experimental study on pulse tube refrigeration with helium and nitrogen mixtures." *Cryogenics* 40.4 (2000): 333-339.
18. Gao, C. M., Y. L. He, and Z. Q. Chen. "Study on a pulse tube cryo-cooler using gas mixture as its working fluid." *Cryogenics* 40.7 (2000): 475-480.
19. Moldenhauer, Stefan, et al. "The pulse tube engine: A numerical and experimental approach on its design, performance, and operating conditions."*Energy* 55 (2013): 703-715.
20. Zhang, X. B., et al. "CFD study of a simple orifice pulse tube cooler."*Cryogenics* 47.5 (2007): 315-321.
21. Antao, Dion Savio, and Bakhtier Farouk. "Experimental and numerical investigations of an orifice type cryogenic pulse tube refrigerator." *Applied Thermal Engineering* 50.1 (2013): 112-123.
22. Yang, L. W., and G. Thummes. "High frequency two-stage pulse tube cryo-cooler with base temperature below 20K." *Cryogenics* 45.2 (2005): 155-159.
23. De Boer, P. C. T. "Performance of the inertance pulse tube." *Cryogenics* 42.3 (2002): 209-221.
24. Charles, I., L. Duband, and A. Ravex. "Permanent flow in low and high frequency pulse tube coolers–experimental results." *Cryogenics* 39.9 (1999): 777-782.
25. Huang, B. J., and T. M. Tzeng. "Performance characteristics of pulse tube refrigerators." *Cryogenics* 33.12 (1993): 1132-1136.
26. Ansys INC.2010. chapter 5.2 User Defined Functions manual.
27. Banjare Y.P, R.K. Sahoo, S.K. Sarangi,"CFD Simulation of Orifice Pulse Tube Refrigerators" International Conference on "Recent trends in Mechanical Engineering (IRCTME) 2007, Dept. of Mech. Engg. Ujjain Engineering College Ujjain October 4-6, 2007, Paper no. HT1, pp235-45.
28. Etaati M.A "Stirling-type pulse-tube refrigerator for 4 K" conference on "Scientific computing and applications", Applied physics Department, CASA, may 5-7, 2007.